AUTOMOBILE ENGINEER

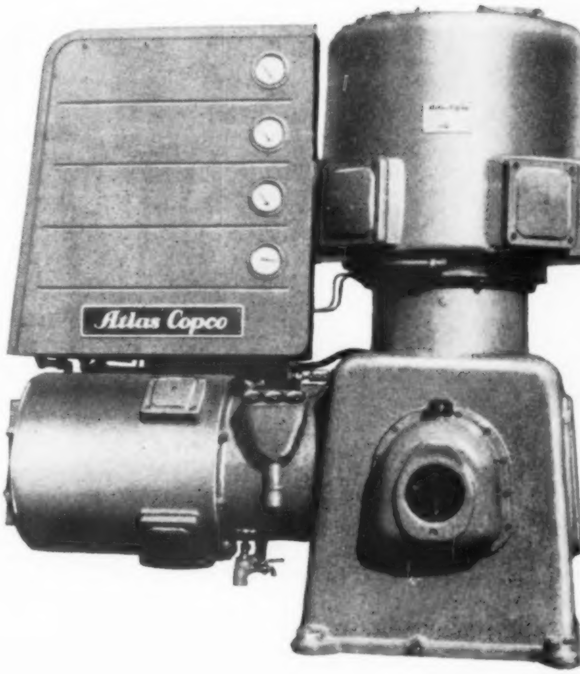
DESIGN · PRODUCTION · MATERIALS

Vol. 50 No. 7 JULY 1960 PRICE: 3s. 6d.

Here's Air Power at

ROCK BOTTOM cost

ER6 Heavy-Duty Compressor for reliable 3-shift operation



Atlas Copco's ER6 compressor delivers air at the lowest possible cost. Designed to incorporate *every* economy factor, it combines the lowest possible running costs with low installation costs and a space saving design.

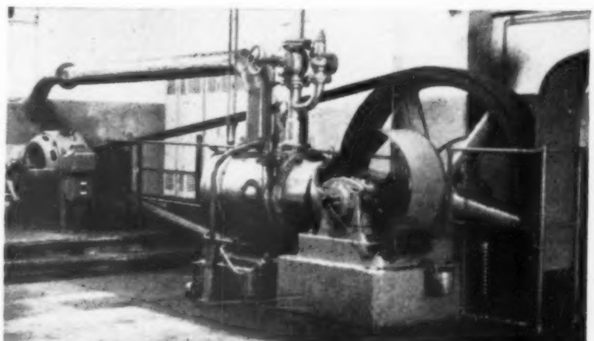
The ER6 delivers 1,075 cfm at 100 psi (30.4 m^3/min at 7 kg/cm^2) for a power consumption that is absolutely rock bottom. Thanks to a new type of intercooler, water consumption is only 440 imperial gallons (2,000 litres) per hour—a *quarter* the consumption of most similar machines. The ER6 is compact. Occupying only half the space usually required for machines of its capacity, it saves expensive factory space. In addition, installation costs are lower as new, cheaper installation methods are employed.

RANGE OF STATIONARIES

From 2.5 to 20,000 cfm, there is an Atlas Copco stationary compressor to suit every requirement. Whatever the design factor — compactness, low installation cost, low operating cost, or a combination of all three — Atlas Copco stationaries are high-performance machines you can rely on.

Sales and Service in Ninety Countries

Represented in ninety countries, Atlas Copco is the world's largest organisation specialising solely in compressed air equipment. Products include stationary and portable compressors, rock drills, loaders, hoists, air tools and paint-spraying equipment. Wherever you are, the international Atlas Copco group offers expert advice and provides a complete after-sales service.



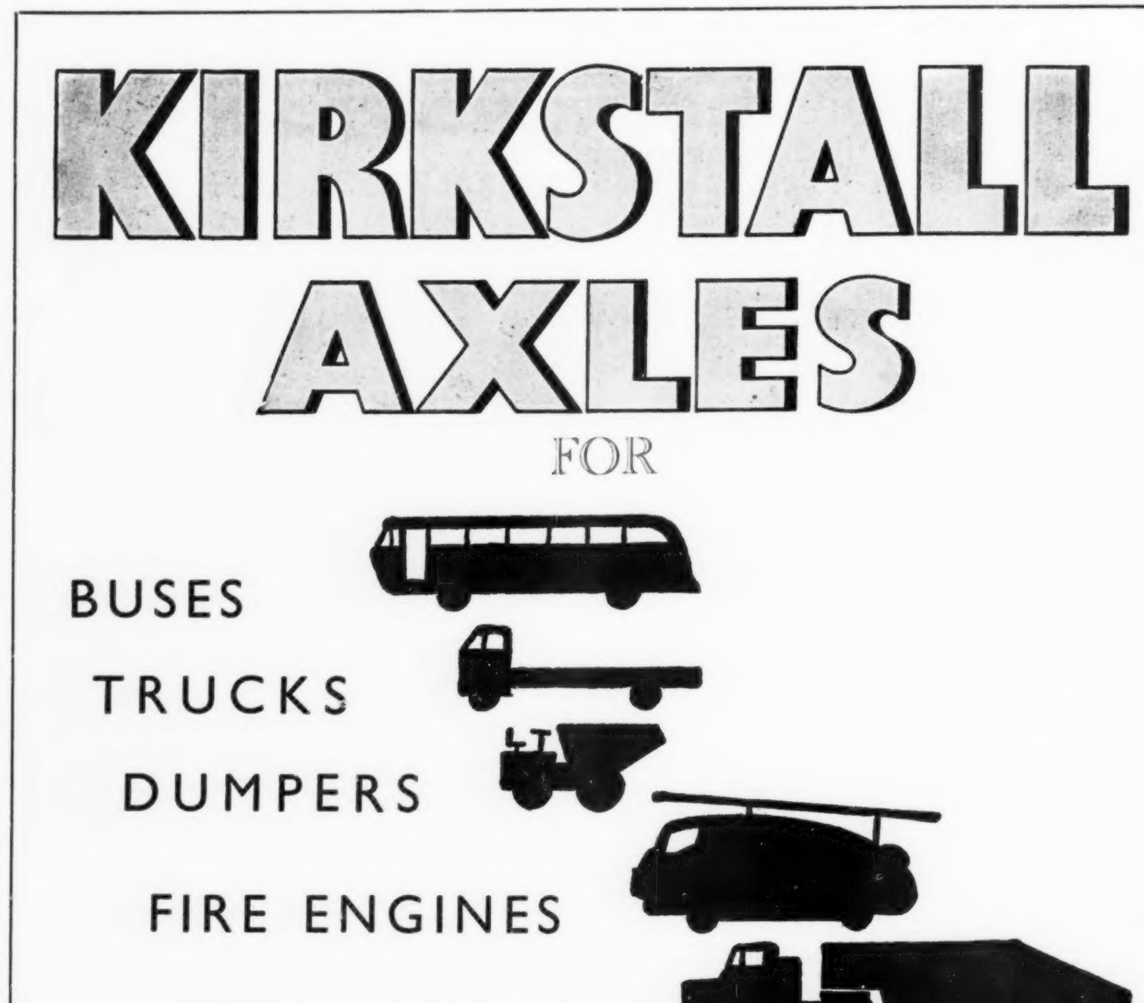
Built to last. This Atlas Copco compressor was installed at the Bofors engineering works (Sweden) in 1907. It is still in operation.

WRITE FOR THE LEAFLET

Leaflet E1127 gives full details of the ER6 compressor. Write for a copy to your local Atlas Copco company or agent or to the address at the foot of this advertisement.

Atlas Copco puts compressed air to work for the world

Atlas Copco AB, Stockholm 1, Sweden. In the U.K. Atlas Copco (Great Britain) Ltd., Hemel Hempstead, Herts. SC 38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MOBILE CRANES
KIRKSTALL FORGE ENGINEERING LIMITED
LEEDS, 5
Telephone: Horsforth 2821

SPECIALIZED COMPONENTS

The Purolator system of filtration will prevent rapid deterioration of tractor engines working under arduous conditions of dust and grit in the field.

AIR FILTERS

The Purolator Dry Type filter is capable of arresting particles of foreign matter small enough to be measured in microns. These filters are particularly suitable for use on combine harvesters where engines work under conditions of excessive dust.

PUROLATOR

Regd. Trade Marks: Purolator, 'Micronic'

MICRONIC FILTERS

OIL FILTERS

Purolator Micronic full flow filters are designed to handle all the oil supplied to the engine bearings under normal operating conditions. A relief valve is incorporated to ensure a continuous supply of oil should the element have been allowed to become choked through neglect. Designed for mounting directly on to a machined facing on the engine crankcase, these models dispense with external piping.

FUEL FILTERS

These diesel engine fuel filters are designed to give final protection immediately prior to the injection pump. The filter bowl has ample capacity for water and sediment which may be drained off periodically.

Our engineers will be pleased to discuss your filtration problems on request.

AUTOMOTIVE PRODUCTS COMPANY LIMITED
LEAMINGTON SPA, WARWICKSHIRE, ENGLAND

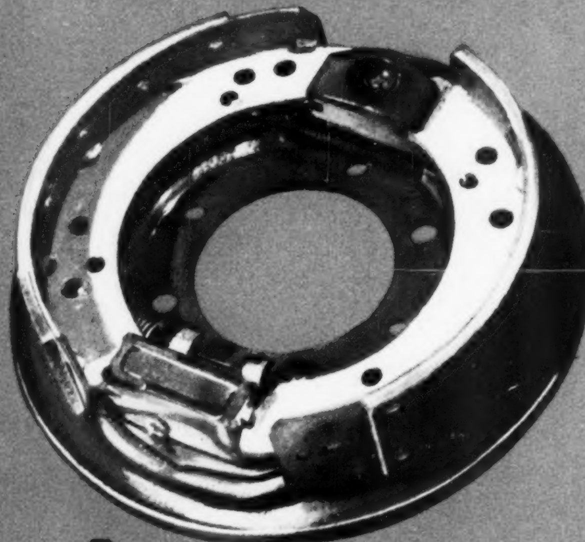
ONE OF THE AUTOMOTIVE PRODUCTS GROUP

Lockheed

REGD. TRADE MARK

MECHANICAL BRAKES

FOR TRACTORS & TRAILERS



For general use as primary brakes and steering brakes for equipment such as farm tractors, farm machinery, farm and industrial trailers, etc., etc.

They form a light, powerful, inexpensive braking system giving controlled safety with maximum economy and negligible maintenance. For actuation manually, or remotely by vacuum and air systems.

Leading and trailing shoe brakes of the centre-pull type, are available in a range of sizes, for 7" drum, shoe width 1¼", 8" (1½"), 9" (1¾" and 2¼"), 10" (1¾"). The 9" x 2¼" brake has a lever for rod or cable operation, to pull across the backplate.

LOCKHEED HYDRAULIC
BRAKE COMPANY LIMITED
LEAMINGTON SPA, WARWICKSHIRE

Lockheed Avery

FLEXIBLE HOSE AND SELF-SEALING COUPLING

The 'Hydralinc' coupling shown enables the user to make the most of his hydraulic implements. Hydraulic lines can be quickly connected and disconnected without loss of fluid or inclusion of air.

There is no need to prime fluid systems fitted with this coupling. For pressures up to 3,000 p.s.i.

Lockheed-Avery hoses are made to a very high specification in order to provide consistent durability in agricultural usage. The end-fittings are re-usable when the hose is renewed.

LOCKHEED PRECISION PRODUCTS COMPANY LTD.
SHAW ROAD, SPEKE, LIVERPOOL, 24.

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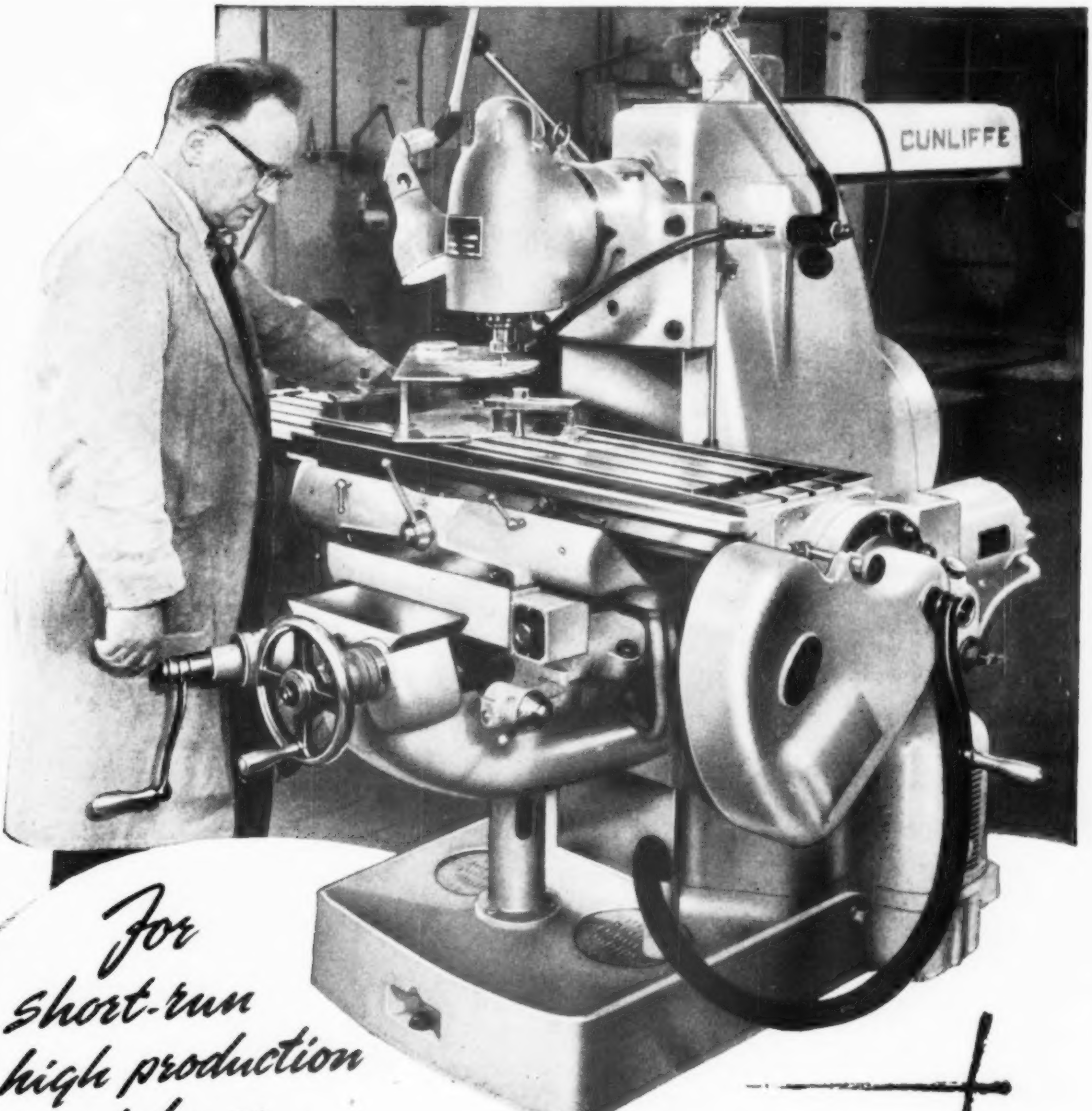

For short-run high production jobs or . .

. . . . or for prototype work as shown, CUNLIFFE & CROOM horizontals, with their in-built features of precision and rigidity, mean increased efficiency wherever they are installed. At the Imperial Tobacco Co., Ltd., Bristol, for instance, this machine, with vertical milling attachment, is fully employed in the Prototype Development Dept., on components for new machines.
CUNLIFFE & CROOM milling machines, horizontal, universal and vertical, are available in a range of sizes to reduce costs on your own work. Complete catalogues available. Write for your copies to-day.

CUNLIFFE & CROOM

HORIZONTAL MILLERS

FULL PARTICULARS OF THESE MACHINES FROM

JAMES ARCHDALE & CO. LTD.

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Works: BLACKPOLE WORKS, WORCESTER. Phone: Worcester 27081 (6 lines)
A Member of the Staveley Coal & Iron Co. Limited Group
Sole Agents: ALFRED HERBERT LTD. COVENTRY Telephone No. Coventry 89221

"Maybe you could express it a little better . . ."

"Well, a chuck is just a chuck unless it is a chuck made by Jacobs when it is a genuine Jacobs chuck the best known name in chucks!"

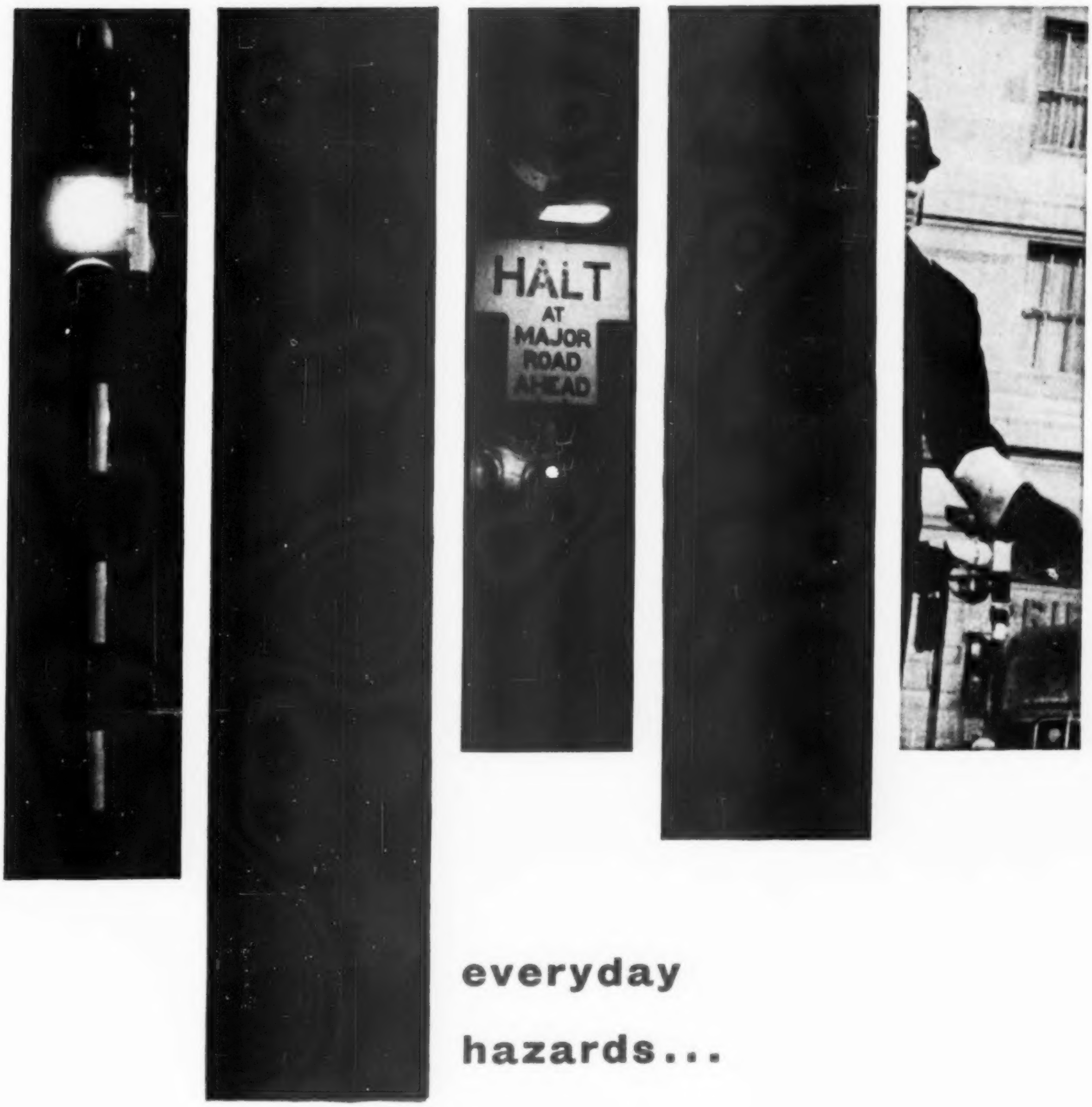
"Your dealer can supply genuine Jacobs chucks in all sizes for light medium or heavy duty tools and machines"

INSIST ON GENUINE

Jacobs

CHUCKS

THE JACOBS MANUFACTURING COMPANY LIMITED, ARCHER RD., SHEFFIELD, 8

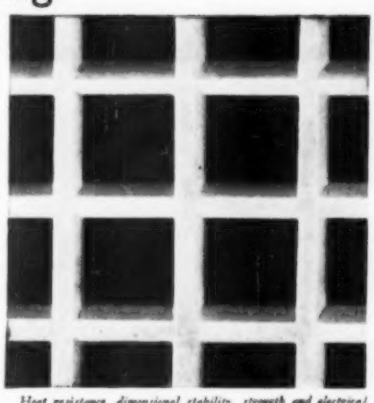

HALT
AT
MAJOR
ROAD
AHEAD

for heat resistance

special "thin-flash" type for grille moulding

Beetle D.M.C. is the material used for the grille of the Morphy-Richards Fan Heater because it offers high heat resistance and good appearance. The use of D.M.C. posed a flash-removal problem in relation to the many apertures, as the moulders, Crystalate (Mouldings) Ltd., specified a very thin flash line. Accordingly, B.I.P. Chemicals Ltd. developed a special type of material (D.M.C. L.1834) which, in combination with a tool made to close limits, ensures a consistently thin and easily removed flash.

Beetle D.M.C. combines heat resistance with strength, dimensional stability and excellent insulation properties. The development of a special type for a single application typifies the extent of B.I.P. services in relation to individual problems and requirements.



Heat resistance, dimensional stability, strength and electrical insulation properties are among the features making D.M.C. the ideal material for this grille. Moulding by Crystalate (Mouldings) Ltd., Tonbridge, Kent.

BEETLE*

Polyester dough moulding Compound

B.I.P. Chemicals Limited, Oldbury, Birmingham. Phone BROadwell 2061
London Office: Haymarket House, 28 Haymarket, S.W.1. Phone TRAfalgar 3121

an extended range of

ASTON FILLER CAPS

by Enots

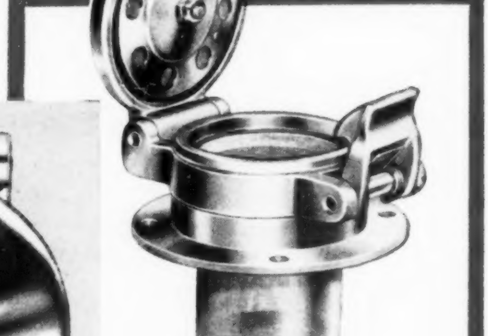
The range of Quick-Release non-ferrous Aston Filler Caps in two standard sizes, now includes a pressurised version to hold up to 10 p.s.i. One of the main design features of this type of filler cap is that the cap itself is hinged and consequently can never become detached and mislaid.

Caps can be supplied with coloured name or trade mark badges. Standard badges include Fuel Oil, Heater Oil, Hyd. Oil, Lub. Oil, Petrol and Water. Special badges can be supplied in reasonable quantities.

now covering a wide field of application in Industry

BREATHER TYPE ASTON

For use with Hydraulic Fluid Tanks and incorporating a gauze breather plate inside the lid. All flange-mounted caps as illustrated below can be supplied with drop-in gauze filters if required.



An efficient, attractive and robust filler cap can improve the value of your product to your customer. Aston Filler Caps can be found fitted as standard to many commercial vehicles, diesel locomotives and some aircraft. The majority of English and many Continental racing cars also fit them. Illustrated list No. AFC. 1059 will give you further particulars — may we send you a copy ?

BENTON & STONE LIMITED · ASTON BROOK STREET · BIRMINGHAM 6
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STOP OZONE ATTACK

WITH DU PONT ELASTOMERS

Ozone is present in the atmosphere, generated from Oxygen in the air by ultraviolet light (from sunlight). When natural rubber is stressed, as in car components, ozone attacks, forming a myriad of small cracks in the surface of the rubber.

DU PONT NEOPRENE has outstanding resistance to ozone cracking. Du Pont neoprene has proved under test, and on the job, that it is vastly superior to natural rubber—and to "general purpose" synthetics. Neoprene resists oil and grease in the chassis . . . high temperatures under the bonnet . . . oxidization and aging of exposed parts and accessories. Properly compounded neoprene products meet most service needs at temperatures ranging from —54°C to as high as 121°C. Compounds can be made soft or hard and in sponge form. **HYPALON*** synthetic rubber has even greater heat and ozone resistance and is known for its remarkable colour stability, permitting use of bright colours for weather stripping and convertible tops. **Hypalon doesn't become brittle in cold weather nor will it crack under heat or summer sun.**

*HYPALON IS DU PONT'S REGISTERED TRADEMARK FOR ONE OF ITS ELASTOMERS
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DU PONT
ESTABLISHED 1802

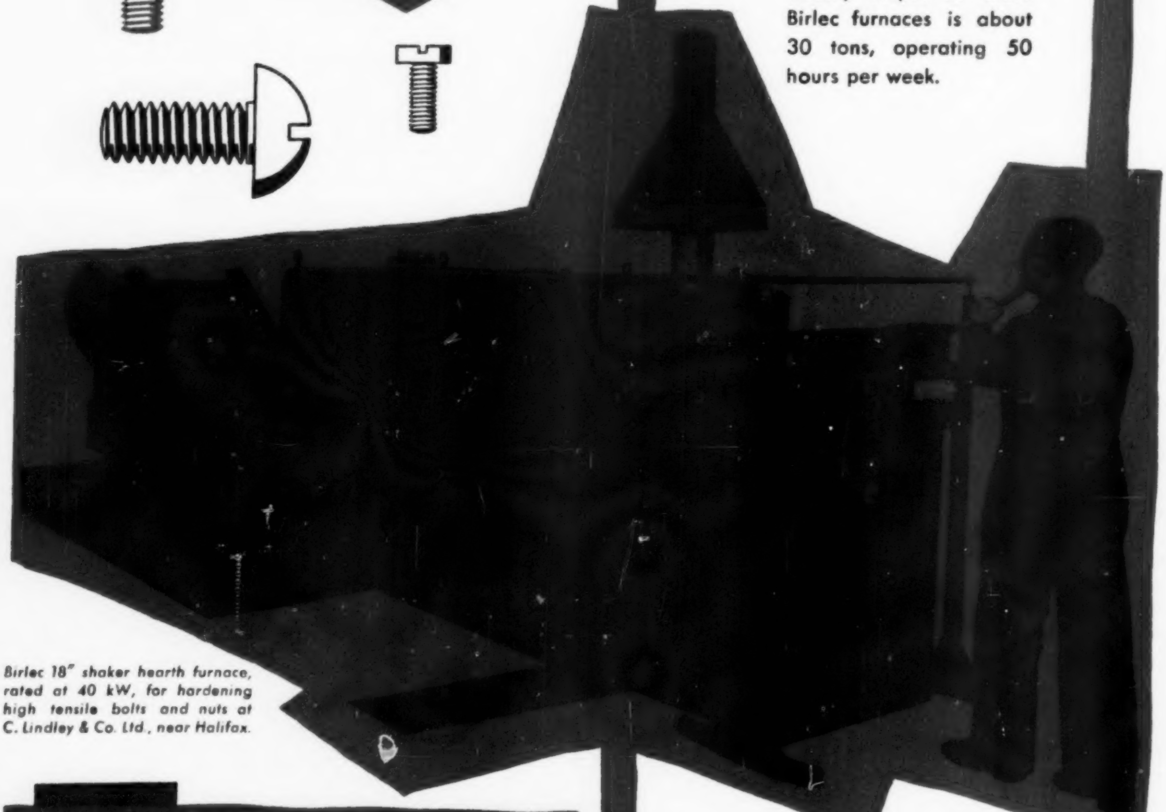
NEOPRENE-HYPALON*

BETTER THINGS FOR BETTER LIVING . . . *THROUGH CHEMISTRY*

(21 765 01)

SECURED BY A THREAD

Whether it be a Whit., BSF, UNF, UNC, or B.A. thread it is all the same to C. Lindley & Co. Ltd., Luddenden, Yorkshire. They make all types of standard high tensile bolts and nuts from 4 B.A. to ⅝" diameter and are well known for the manufacture of special parts by the cold heading method. Their heat treatment, which involves hardening tempering and normalising, is entirely carried out in four Birlec atmosphere controlled furnaces including 18" and 24" Birlec shaker hearths, and Birlec cast link conveyors. The weekly output of these Birlec furnaces is about 30 tons, operating 50 hours per week.



Birlec 18" shaker hearth furnace, rated at 40 kW, for hardening high tensile bolts and nuts at C. Lindley & Co. Ltd., near Halifax.

EXTRA-SPECIFICATION FURNACES *

** The cost of a furnace is not necessarily its purchase price. Even a brief interruption in production may cause severe losses in output and serious inconvenience. The purchaser of a Birlec furnace can be confident that the equipment will not only meet his specification, but will give uninterrupted trouble-free service.*

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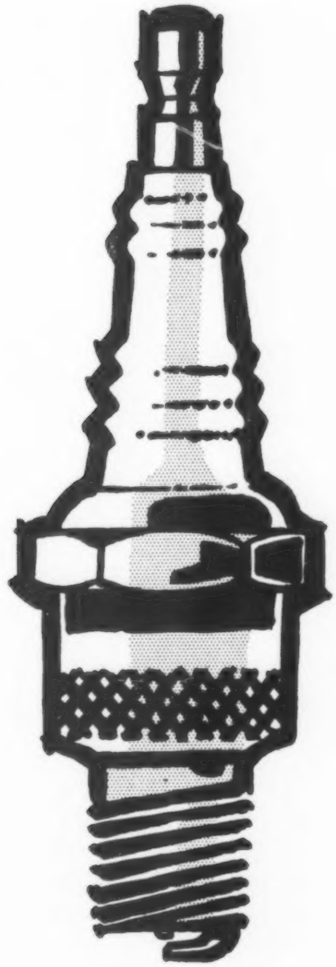


when the going is tough . . . .

DROP FORGINGS

by

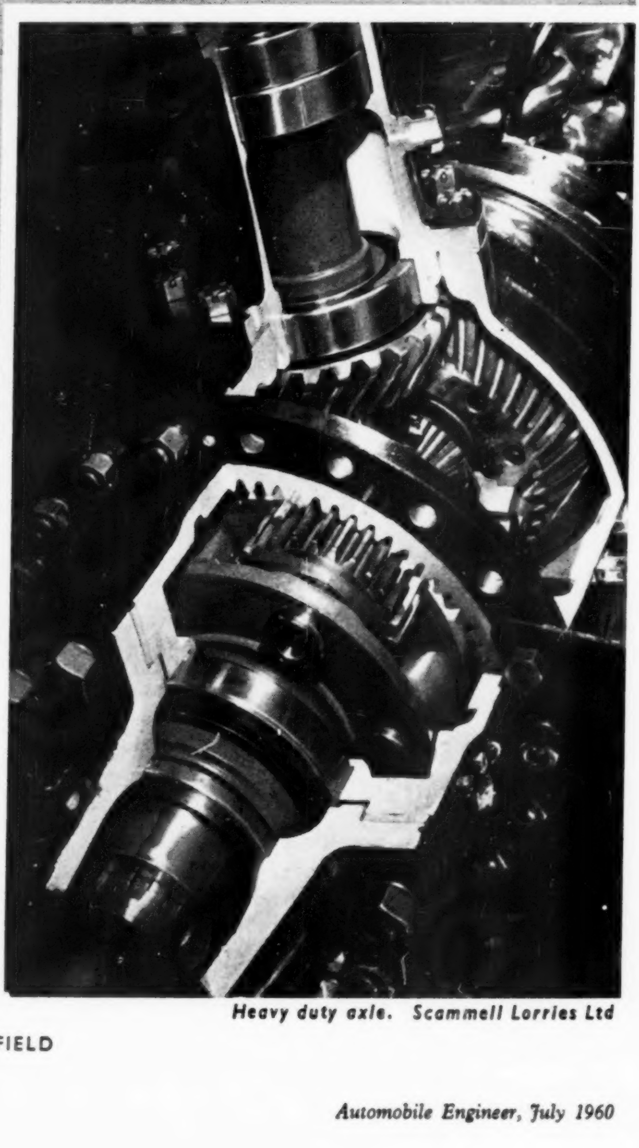
FIRTH-DERIHON

SHEFFIELD

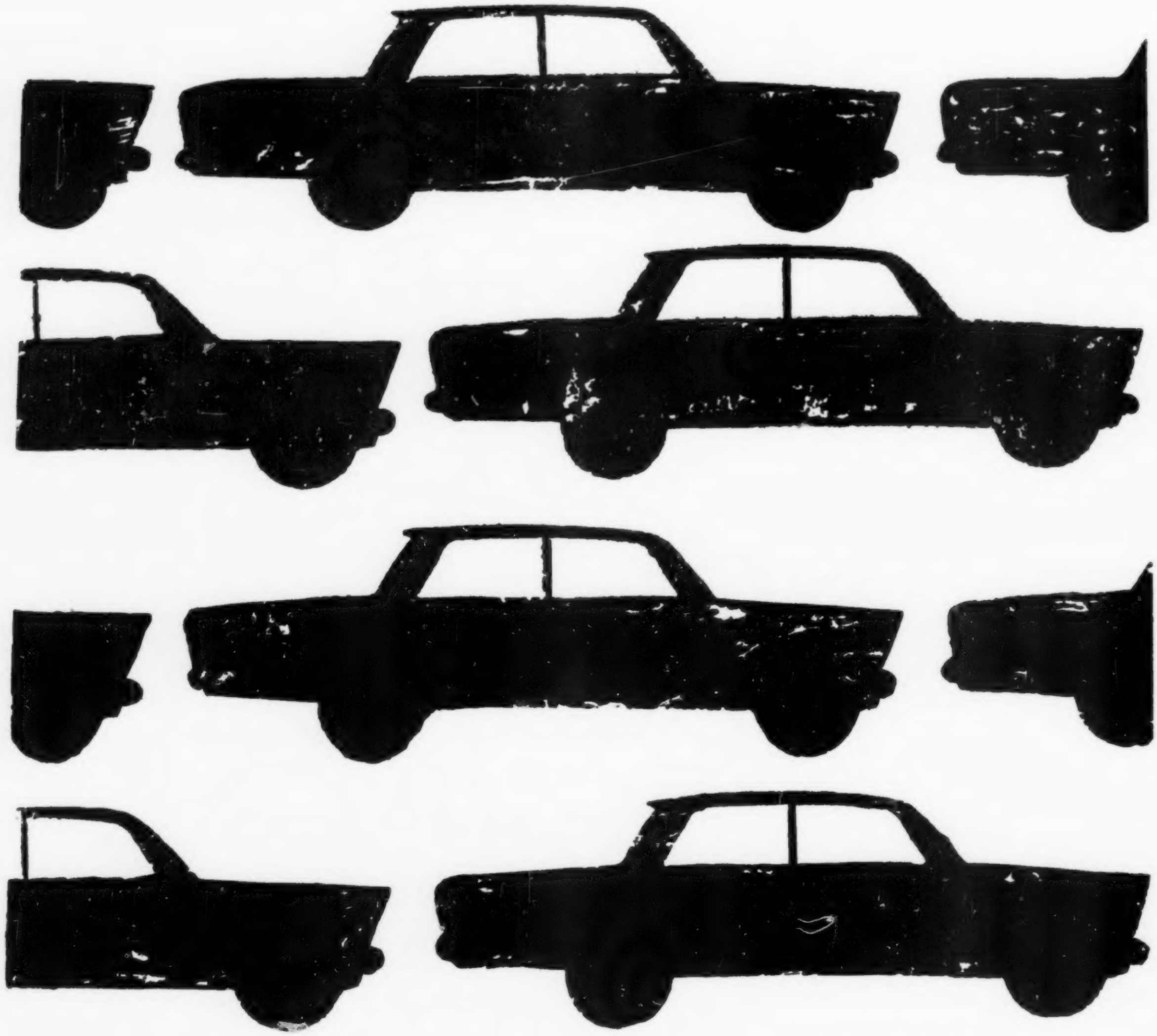
& DARLEY DALE

A 16mm. Colour Film with sound commentary, entitled "Drop Forgings in Alloy Steels," is available on request.

THE FIRTH-DERIHON STAMPINGS LIMITED, SHEFFIELD



Heavy duty axle. Scammell Lorries Ltd



LATER MODELS—LIGHTER METALS Today's outstanding designs are distinguished by bolder use of aluminium in body, engine, trim and detail. Engine designers know all about the advantages of aluminium, while modern alloys and techniques offer new opportunities for far reaching improvement in construction. **IMPALCO** supplies aluminium in many forms from ingot to specially wrought products for individual industries: offers, too, the combined technical resources of its parent companies, ICI and the Aluminum Company of America.

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Imperial Aluminium Company Limited · Birmingham


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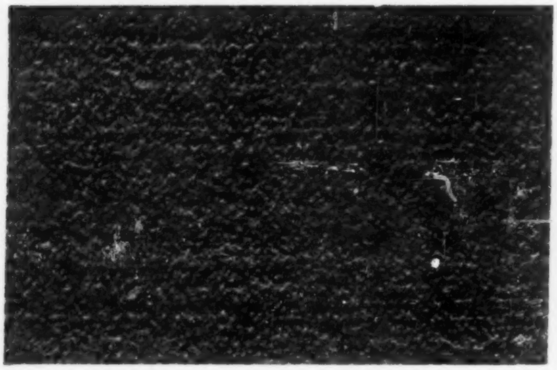
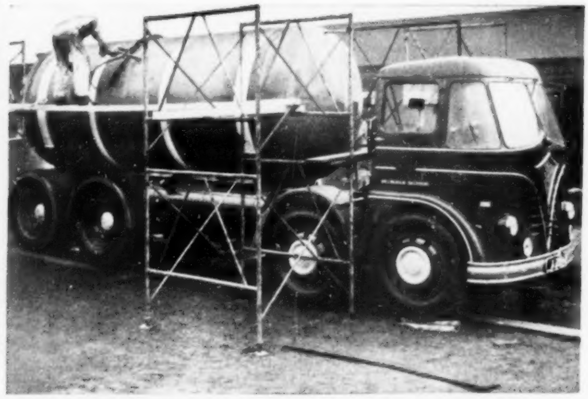
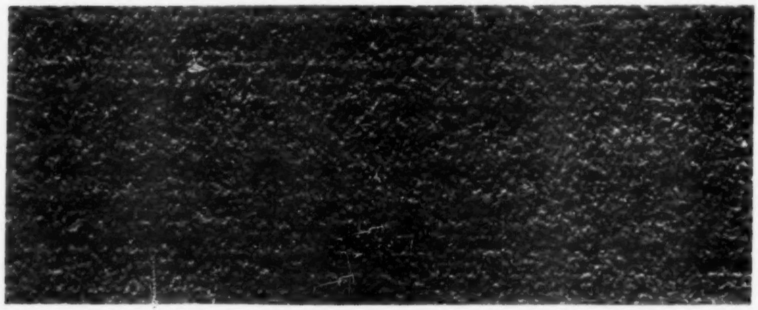





The greatest variety of COATED SHEETS

Richard Thomas & Baldwins, pioneers of modern steel sheet manufacture, offer the widest variety of coated sheets.

RTB hot-dipped and electrolytic tinplate are world-renowned; so are RTB heavily-tinned sheets, used for a wide range of applications from gas meters to dairy utensils.

RTB 'Speltafast' galvanized sheets (flat and corrugated) and coiled strip, made in the most up-to-date plant of its kind, retain their tight coating of zinc spelter, which stands up to pressing and forming without flaking.

There are RTB coated sheets FOR EVERY MANUFACTURING PURPOSE

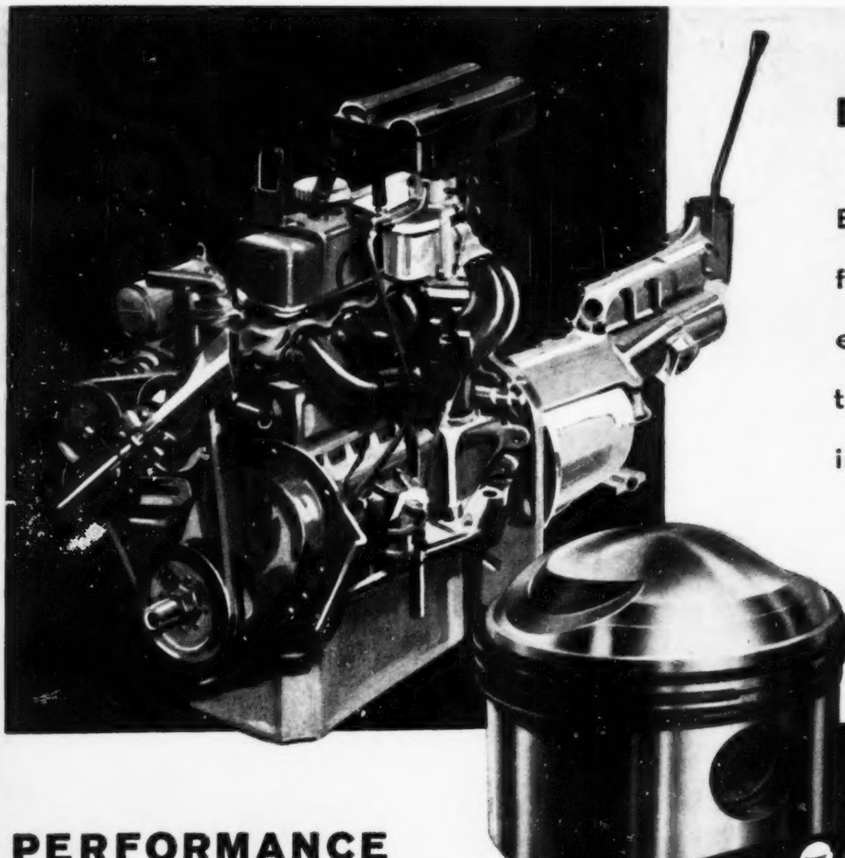
For example, RTB tin-terne, terne-coated and lead-coated sheets, all having a high corrosion resistance and lending themselves to easy fabrication, are used for a great variety of finished products ranging from petrol tanks to ventilation ducting.

A recent RTB development is the 'ARTBRITE' series of P.V.C. coated sheets, available in various decorative finishes, glossy and matt, plain and patterned. Normally based on RTB galvanized sheets, 'Artbrite' sheets are resistant to abrasion and to a great many chemicals; further, they have extraordinary 'workability', being suitable for practically every operation that the RTB steel sheets will stand. 'Artbrite' sheets are weather-resistant.

Richard Thomas & Baldwins (Sales) Limited

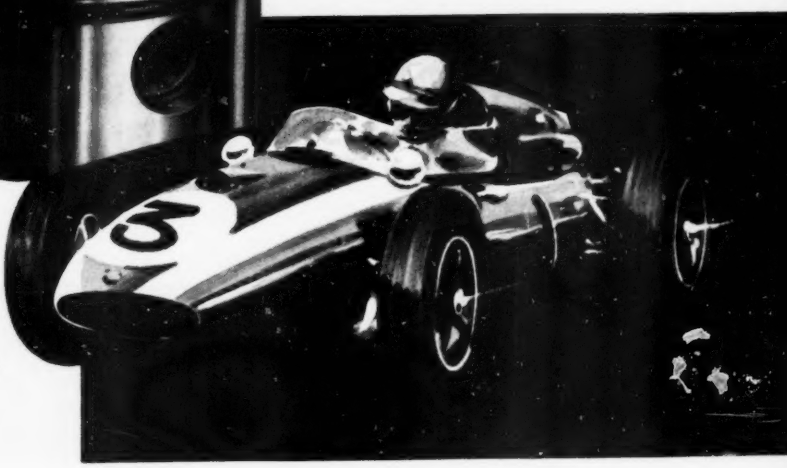
HEAD OFFICE: 47 PARK STREET, LONDON, W.1

RTB



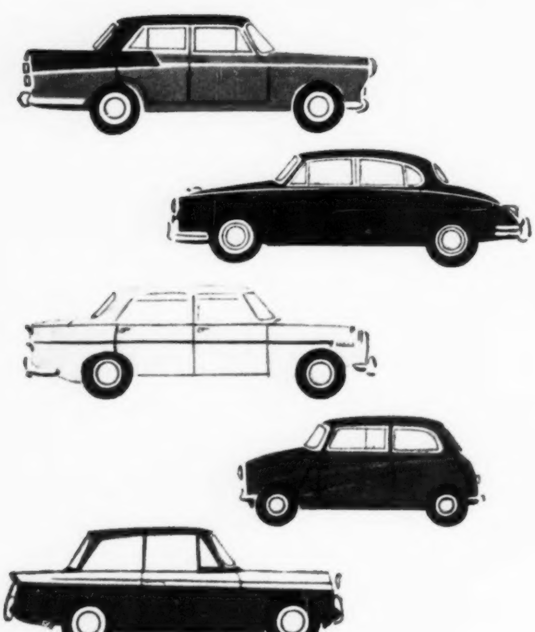
PRODUCTION

BRICO make all the pistons for the famous "A" series engines for B.M.C. vehicles—the biggest piston job in the country.



PERFORMANCE

BRICO make the pistons for the Coventry Climax $2\frac{1}{2}$ litre engine—champion Grand Prix engine of 1959.



POPULARITY

BRICO make the pistons for the following world-famous firms: AUSTIN · BRISTOL-SIDDELEY · COVENTRY CLIMAX · HUMBER · JAGUAR MORRIS · ROVER · STANDARD

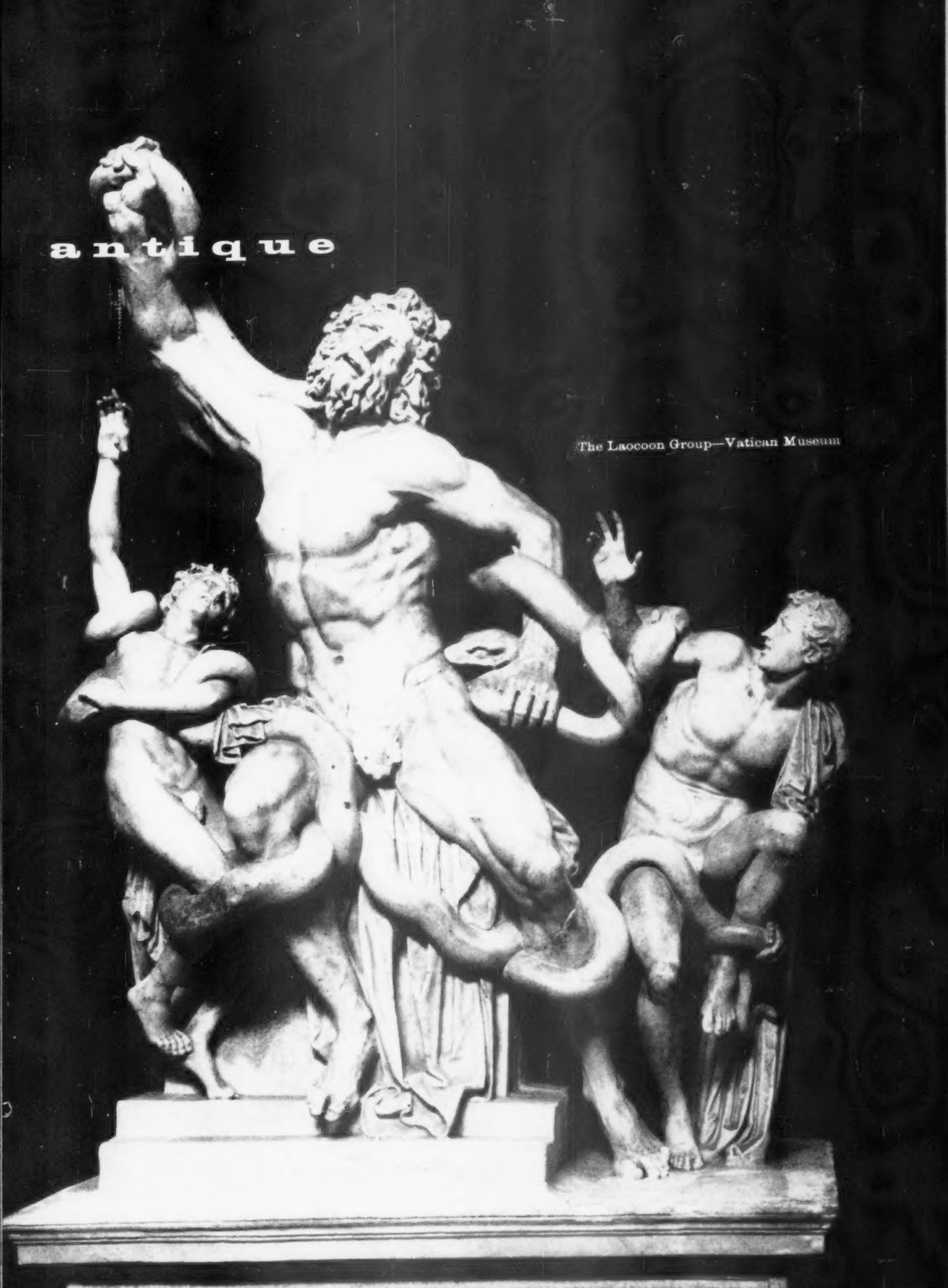
TRADE MARK

THE BRITISH PISTON RING CO. LTD.,
COVENTRY, ENGLAND

601 Cogent

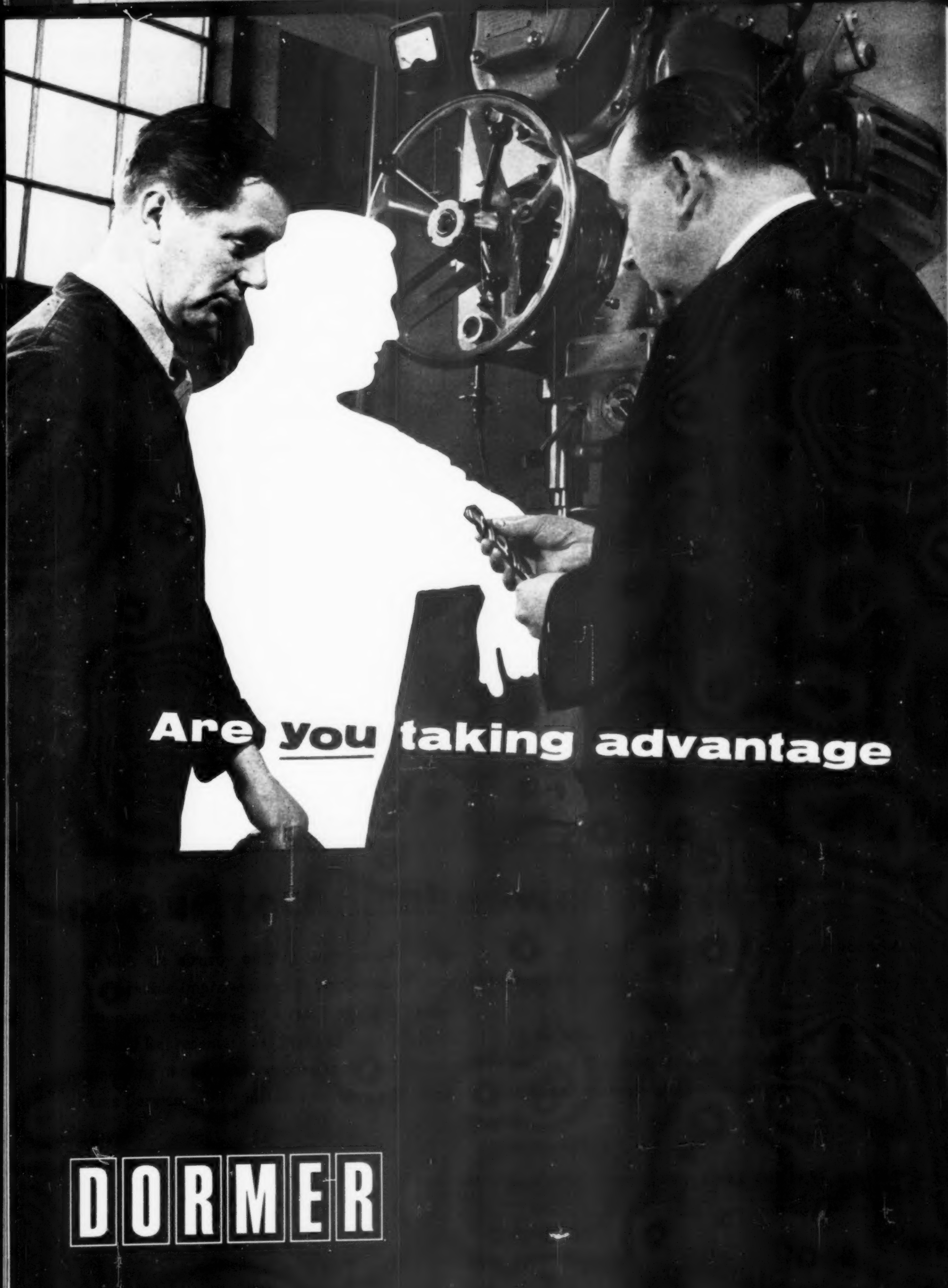
Is it almost impossible?
H.D.A. are unlikely to be disconcerted
The making of these components demanded a material combining outstan formability
with great strength. Fortunately for all concerned, the manufacturer took problem
to H.D.A., who evolved special treatments for one of their Hiduminium Alloys which
gave it exceptional properties for deep pressing and spinning operation
Out-of-the-way problems of this sort are, as you may suspect, only to common.
May we suggest that it is better to take them to H.D.A. at once rather eventually?
H.D.A.'s sleeves are always rolled up. Their thinking-cap is always on.
Hiduminium
most of Aluminium
HIGH DUTY ALLOYS
CKS.
Spinnings & pressings
by Argonarc

antique



The Laocoon Group—Vatican Museum


WITHSTANDS HIGH PRESSURE
Oilite
WEAR RESISTING
SHOCK RESISTING
Oilite
SELF-LUBRICATING BEARINGS AND POWDER METAL PARTS
THE MANGANESE BRONZE & BRASS CO LTD Elton Park Works Hadleigh Rd Ipswich · Grams: Oilite Ipswich · Tel: Ipswich 55926


Are you taking advantage
DORMER

The "Margam Abbey"—built for the Port Talbot Pilotage Authority by Richard Ironworks Ltd., in which Cor-Ten was used extensively. Photograph by courtesy of the Port Talbot Pilotage Authority

COR-TEN GOES DOWN TO THE SEA IN SHIPS

At sea as well as ashore new uses are constantly being found for SCW Cor-Ten. It was used extensively in the construction of this vessel for the Port Talbot Pilotage Authority.

The Main Deck and Bulwark Plating
Shell Plating and Anchor Recess
Floors and Engine Seating
Chain Locker
Fresh Water Tank
Bulkhead Bottom Plating
Rudder Side Plates

—were made entirely from Cor-Ten. A typical example of the growing use of Cor-Ten in marine construction, and in other fields where high yield strength, together with outstanding resistance to corrosion, is vital.

COR-TEN IS TOUGHER

Weight for weight, the yield strength of SCW Cor-Ten is 50% higher than ordinary mild steel.

Alternatively—
Strength for strength, a saving of 1/3 of the weight is possible.

4-6 times more resistant to atmospheric corrosion. Highly resistant to abrasion and fatigue.

COR-TEN SAVES MONEY

Initial costs are spread over a longer service life.

Maintenance costs are reduced.

Operating costs are lowered—in transport, payloads are bigger because of reduction in tare weight.

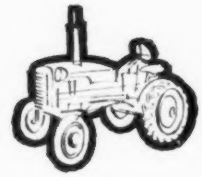
Please write to us at the address below for further information or for technical assistance in the application of SCW Cor-Ten to your products.

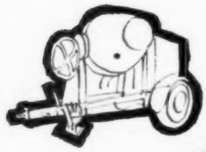
S·C·W BRAND
Cor-Ten

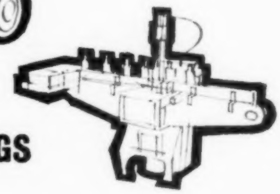
RAILWAY ROLLING STOCK. AGRICULTURAL AND EARTH-MOVING EQUIPMENT. MINE CARS. POWER STATIONS INSTALLATIONS. BARGES AND SMALL CRAFT

THE STEEL COMPANY OF WALES LIMITED

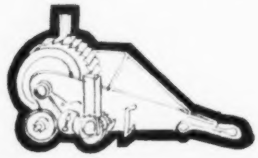
ABBEY WORKS, PORT TALBOT, GLAMORGAN. TELEPHONE: PORT TALBOT 3161

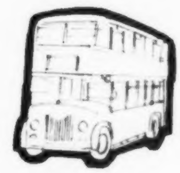






SPEED YOUR PRODUCTION with QUALITY CASTINGS





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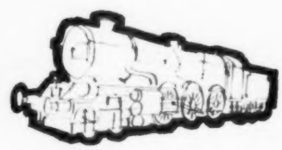
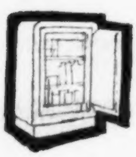
PEARLITIC MALLEABLE
WELDABLE MALLEABLE

Sharp and accurate, designed and produced to save up to 10% in your production costs



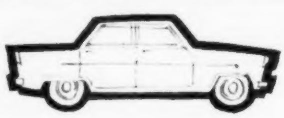


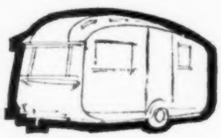
CASTINGS LIMITED

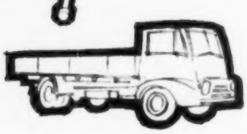
WALSALL, STAFFS
MALLEABLE IRONFOUNDERS
FOUNDED 1835

Office & Works Selborne Street, Walsall
Lichfield Road, Brownhills

Telephone: Walsall 3118-9 Brownhills 2318
Telegrams: Castings, Walsall







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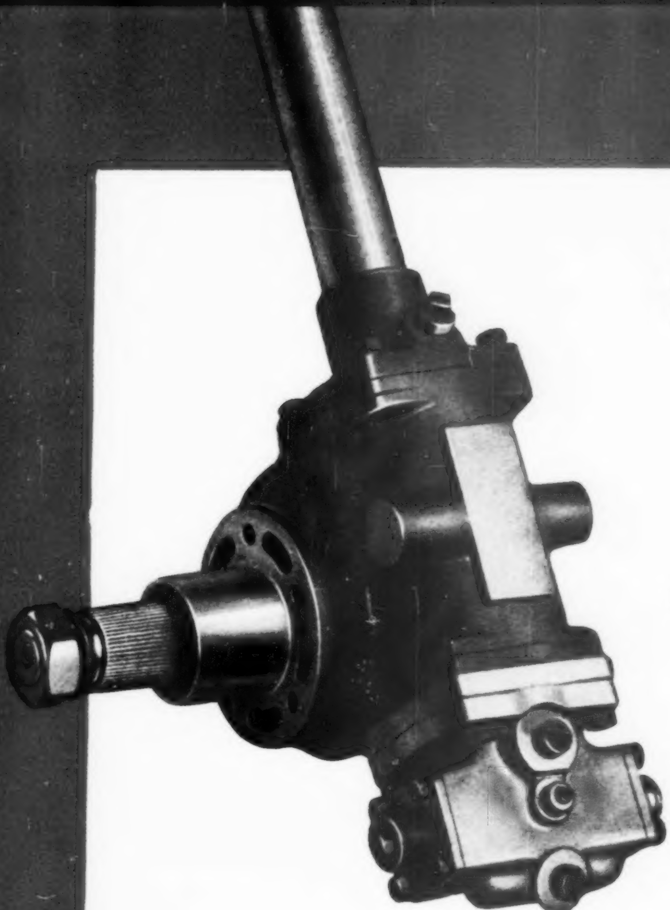

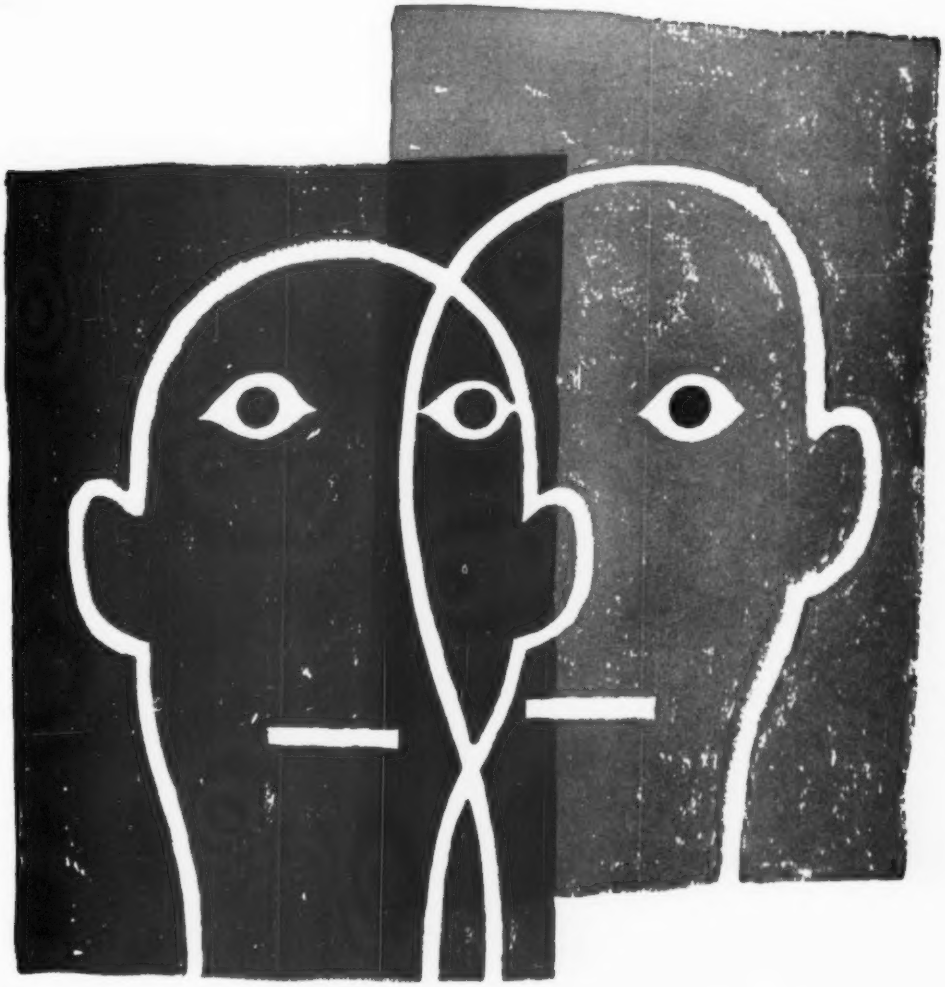

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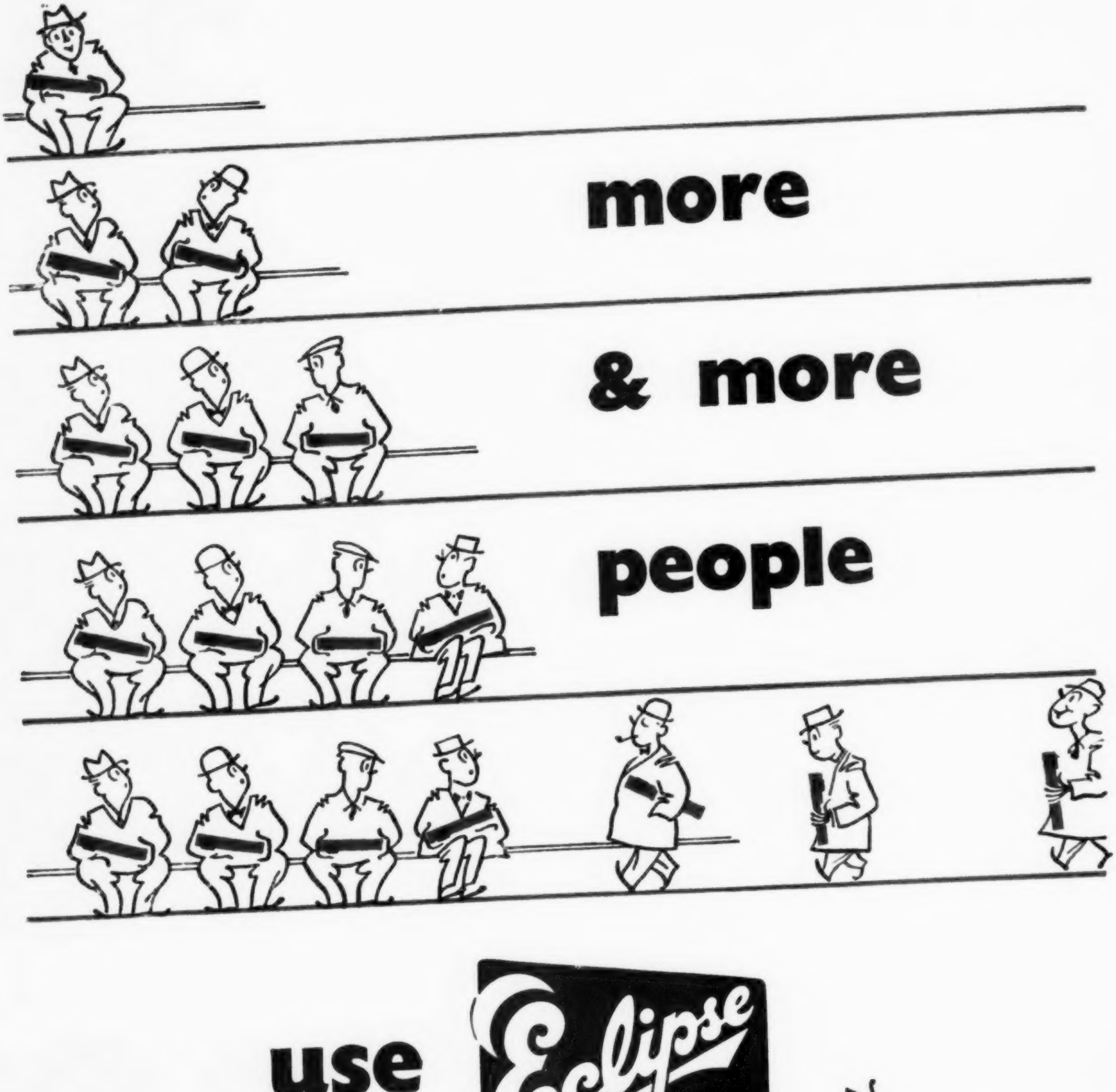

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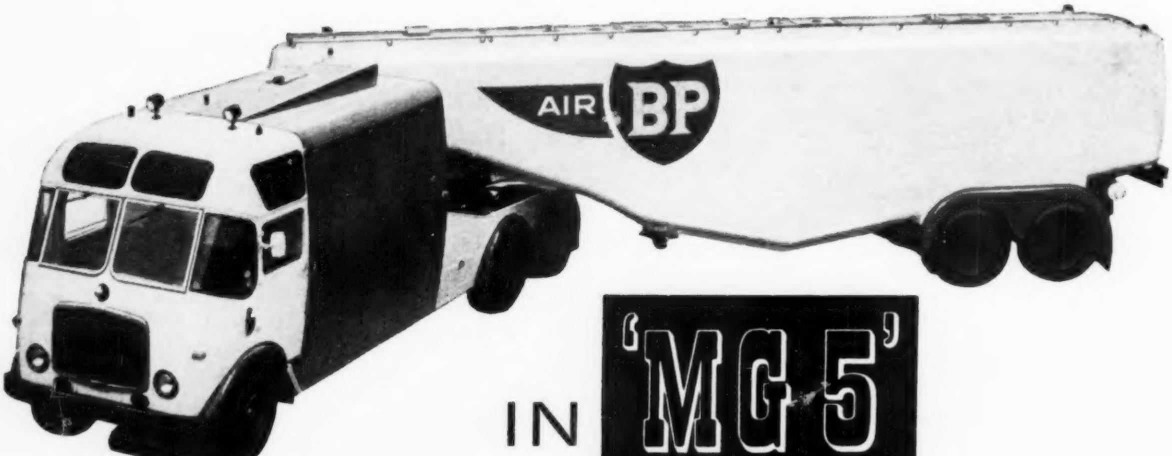

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LEFT *The Super Fueller is powered by a 165 b.h.p. diesel engine which is also used to pump the aviation fuel at flow rates of over 2 tons per minute. Pumping and metering equipment are housed in the compartment behind the vehicle's cab.*

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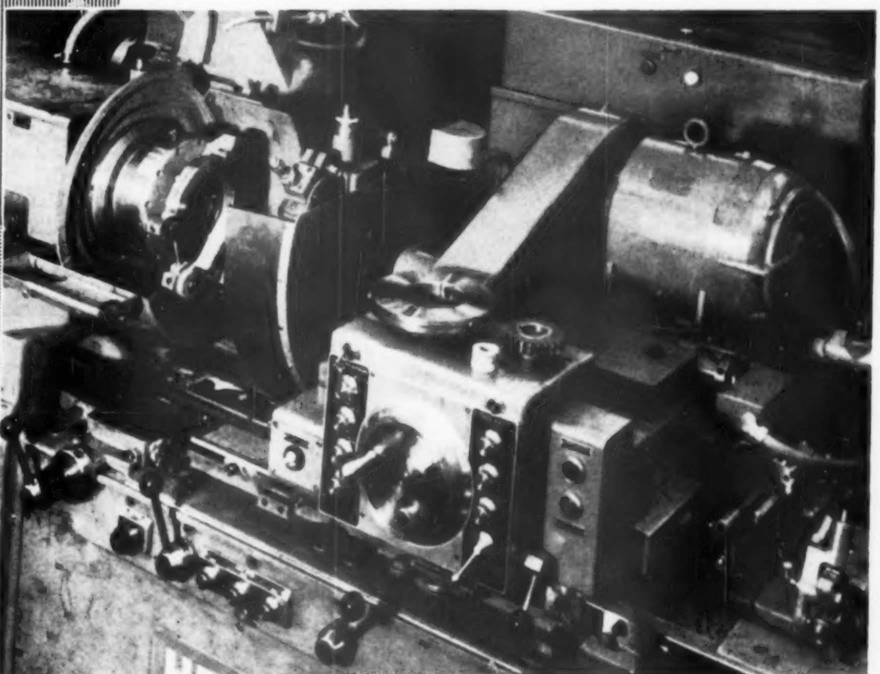

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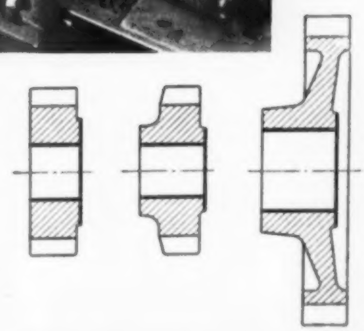

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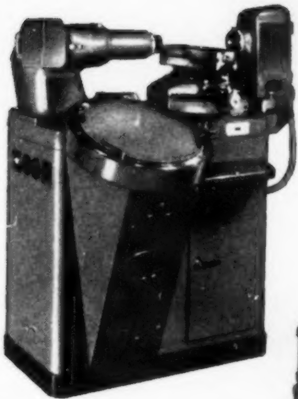

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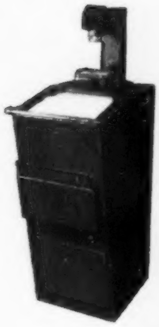

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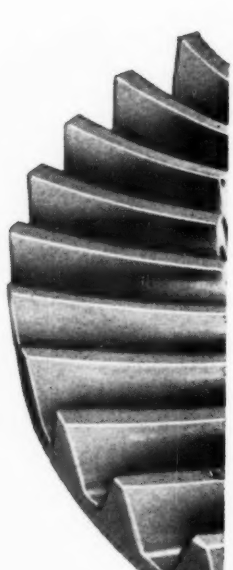

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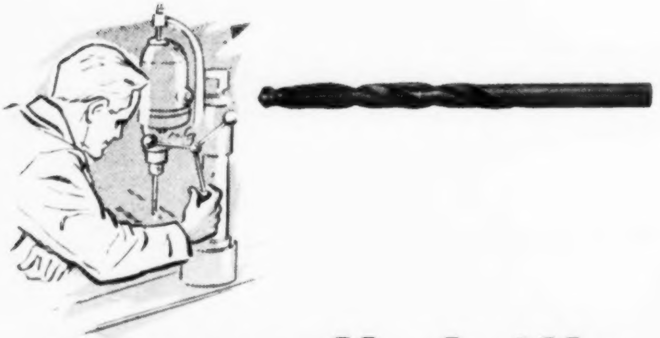

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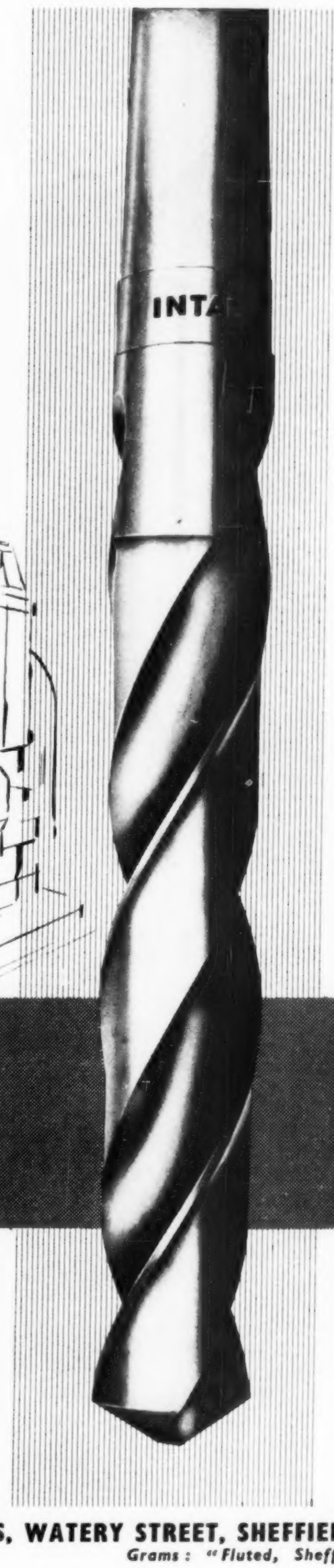

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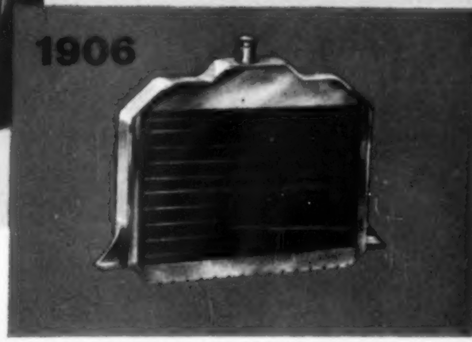

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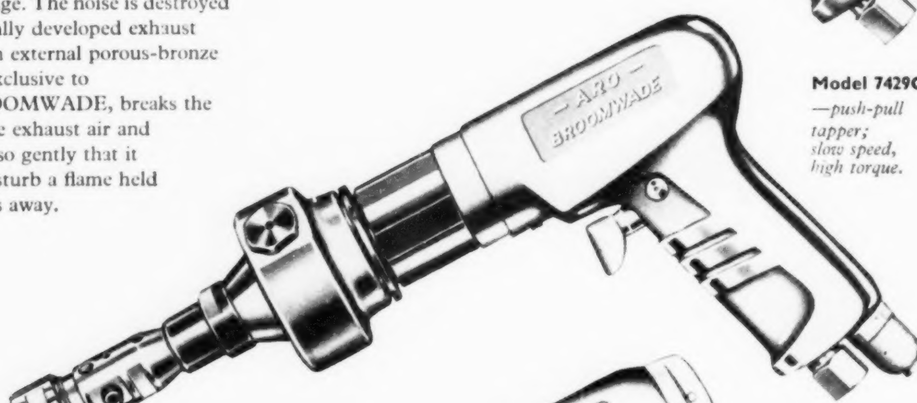

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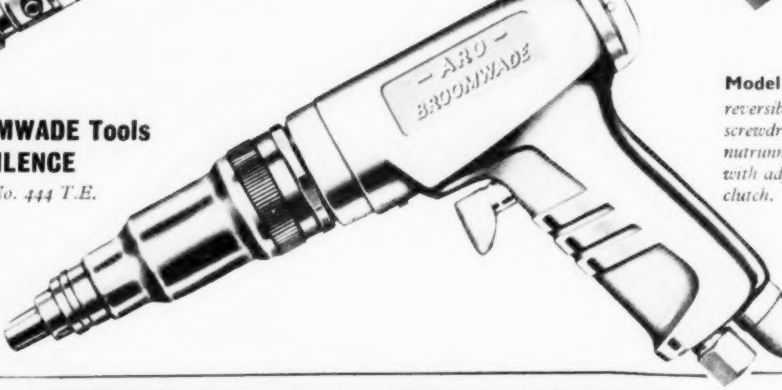

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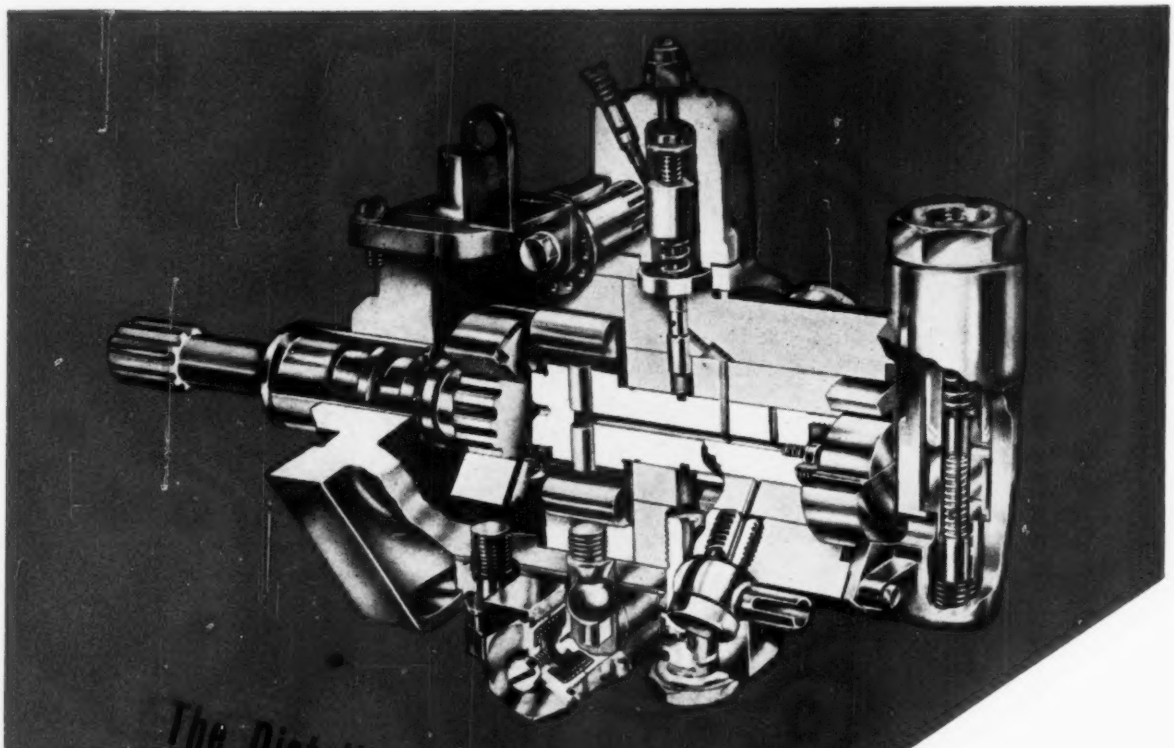
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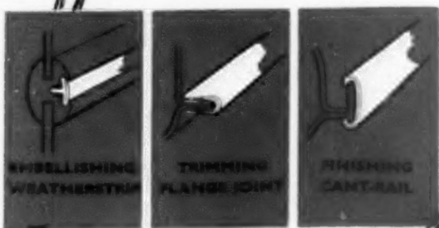

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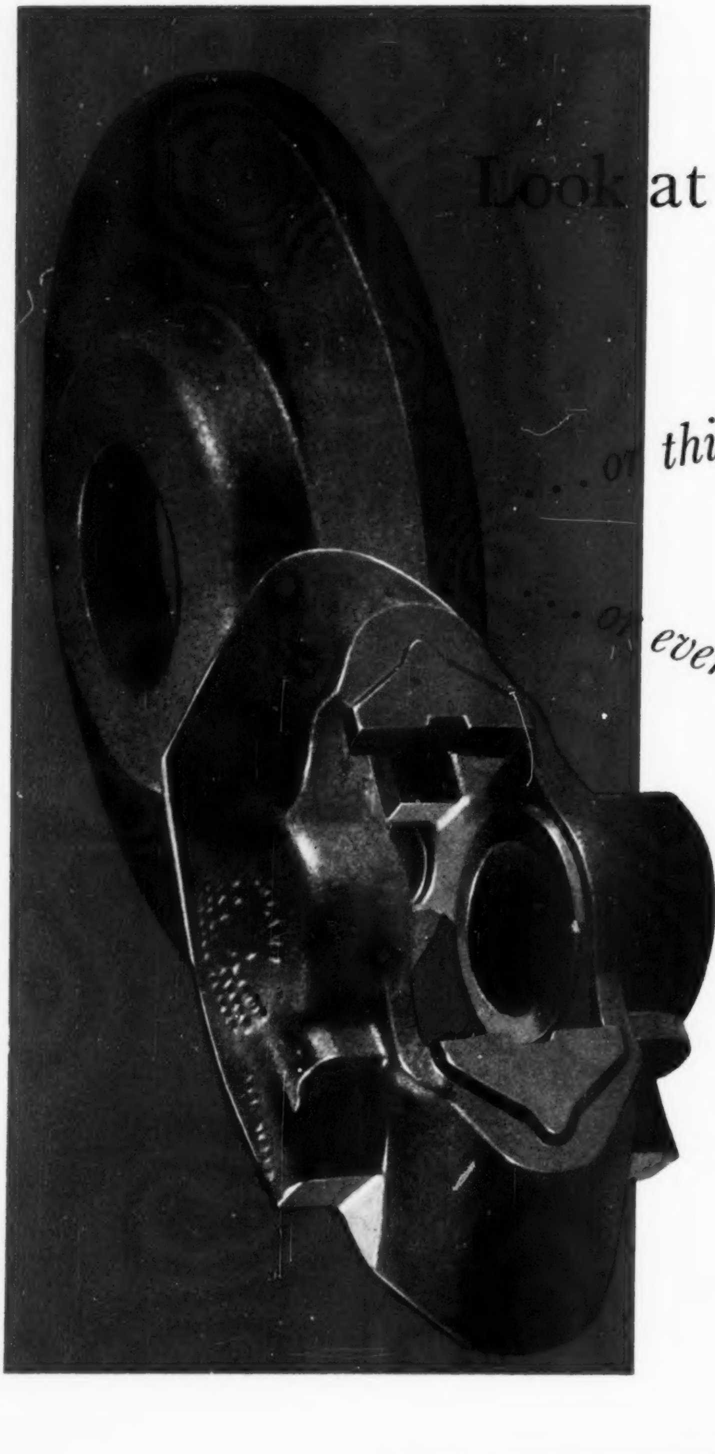

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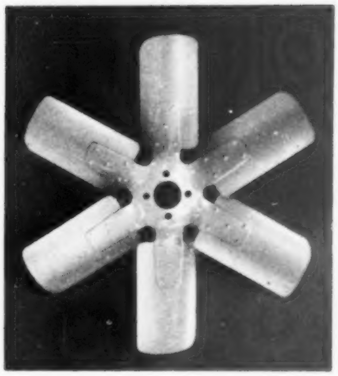

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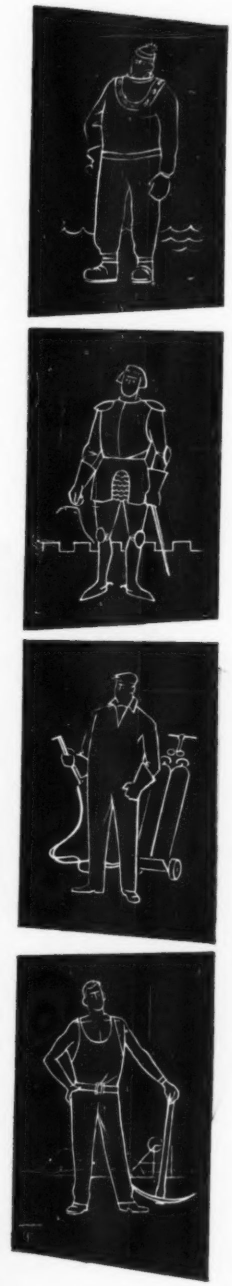

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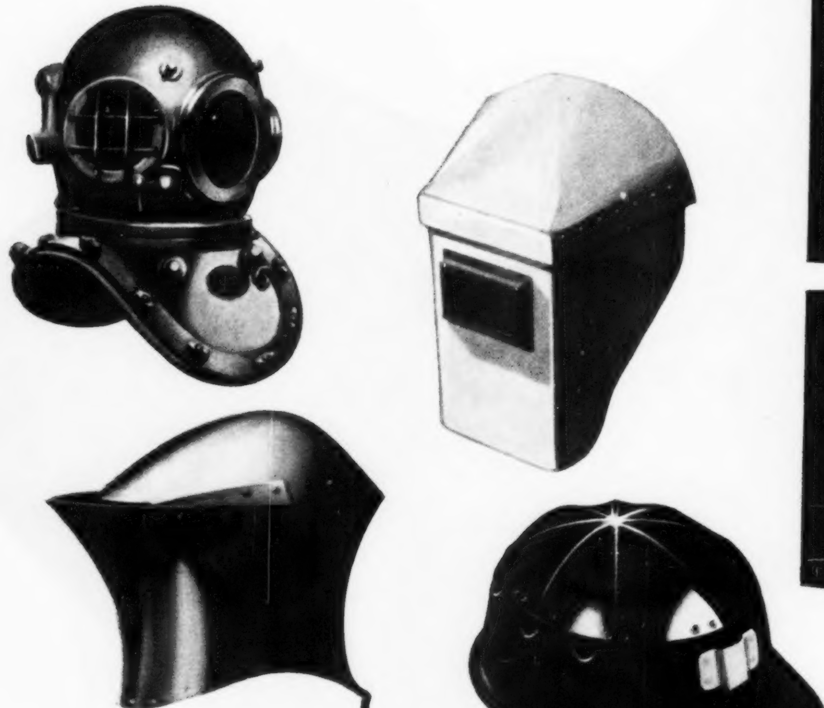

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Non-Metallic Materials

ALTHOUGH, so far as the performance of their mechanical assemblies is concerned, motor vehicles have attained a relatively high standard, there remains a great deal to be done to improve the finish of the body components, especially with regard to durability. Since the latter quality is largely dependent upon the quality of paints, and rubber and other plastics materials, it is perhaps understandable that development in this respect has lagged behind that of the mechanical components, for whereas engineers are operating on familiar ground when they are dealing with metals, few have a thorough understanding of the properties of non-metallic materials. Complicating the problem even further is the fact that the works chemist tends to talk a different language from the engineer, who therefore does not always appreciate the full significance of his reports.

Non-metallic materials might well be used instead of metallic ones in certain applications. The shortcomings of chromium plate are, of course, well known to engineers, as also are the remedies and their high cost. However, the durability of the finish must be improved, either by accepting the higher cost or by eliminating chromium plate. Currently the trend is to employ alternative materials, but even more might be done in this respect. For example, the use of a bright finisher strip either to conceal or to lighten the appearance of heavy black rubber glazing seals could be avoided by the substitution of one of the newly developed self-coloured elastomeric materials for the unsightly black rubber. Because of recent reductions in price—of as much as 30 per cent in one instance—of some of these new synthetic rubbers, there is a case for reconsidering them for many applications.

On the subject of rubber in general, the appearance of the small components of this material, including sections for glazing seals, grommets round petrol tank filler tubes and sealing rings for lamp bezels, contributes a great deal to the overall finish of the vehicle. Moreover, the tendency of certain types of rubber to stain paintwork ought to be investigated further. Natural rubber is, of course, subject to ozone cracking in stressed regions, and the onset of this trouble can quickly render components unsightly as well as ineffective. What is particularly serious is that cracking can occur during transit, for example by ship, to distant countries and can be evident by the time that the new vehicles are exhibited in overseas distributors' showrooms. As is well known, one remedy is to mould the rubber to the shapes of the components around which it fits, so that to all practical intents it is unstressed. However, this can add to the cost and it is not always possible to stress the material lightly and at the same time seal effectively.

Neoprene is well known for its resistance to deterioration when contaminated with petrol or oil, and also it is not subject to ozone cracking. Producers of this material have recently made considerable efforts to reduce the price of finished sections and components, and one outcome is that an extruded neoprene sponge that is self-skinning as it comes from the die is currently under development. The formation of a skin on the sponge is, of course, a useful feature in that it prevents absorption of moisture and the consequent corrosion of the components with which it is in contact. Since these sections are made in one continuous operation, they should be no more costly, and possibly even cheaper, than the currently employed sponge materials with separately applied skins. Moreover, a straight extruded neoprene section can be used, and bent where necessary to conform with the shapes of the components to which it is applied. This new material could be particularly useful for door seals, especially if the door openings were designed in such a way that seals of uniform section could be used.

Perhaps the most important single factor affecting finish is paint. A great deal of research is being carried out in this field and there is of course scope for more. Deterioration can be due to a number of factors, among which are abrasion caused by dust when the vehicle is cleaned and polished, and also chipping of paintwork by stones flung up from the road by vehicles in front. Among the requirements, obviously, are hardness and toughness: if the chemical problems can be solved, this might be achieved by using epoxide resin in conjunction with the alkyd types currently in use. Epoxide resin is very hard and does not flake easily if the panel on which it is applied is dented. Its disadvantages, however, are high cost and a tendency for the gloss to deteriorate after exposure for long periods, but these might be overcome by further development. With regard to the deterioration of gloss, if the material were used as one of the ingredients of an undercoat, this trouble would not be experienced since the finishing coats would protect it from direct exposure.

Hitherto, the specifications for non-metallic materials have often been formulated more or less as an afterthought and, even then, on a basis of inadequate knowledge of fundamental properties; however, much good work has been done by the S.M.M. and T. Standards Sub-committee on non-metallic materials, and a greater degree of rationalization is now being introduced in respect of performance specifications. Synthetic materials are becoming widely used for essential components of motor vehicles manufactured in the United States, and there is certainly scope for more serious consideration of them in Europe.

Austin-Healey Sprite

Part I: The Unitary Structure of the Two-Seat Open Body; the 948 cm³ Engine; Four-Speed Gearbox and Hypoid Bevel Final Drive Unit

The Sprite has trim lines and a minimum of external decoration; an unusual feature is the head-lamp nacelles. A readily fitted hard top, of reinforced plastics material, is an optional extra

THE Austin-Healey Sprite is the smallest sports two-seater in quantity production by any of the larger British car manufacturers. Its originator was Donald Healey, who designed and built the prototype, as he did in the case of the highly successful, larger Austin-Healey model. After extensive testing by him and by the Austin organization, it was passed, for production, to the Abingdon factory of the British Motor Corporation, where all B.M.C. sports models are built. The first production car was completed in March 1958.

Perhaps the most noteworthy feature of the Sprite, apart from its small size, is its chassisless design, the body shell forming also the structural framework of the car. Although common enough in closed cars, this construction is unusual in open sports models, but the road behaviour of the Sprite and the reputation acquired since its introduction appear to justify completely the choice of structure. For this reason, and also because the mechanical assemblies of the car comprise largely B.M.C. standard components, it is the structural aspect of the Sprite that arouses most technical interest. The rear suspension, however, is also unusual in that it embodies quarter-elliptic springs.

The body structure

There are three distinct stages in the building of the Sprite body. The first of these is the production of the under-frame unit by John Thompson Motor Pressings Ltd., at Wolverhampton. This assembly incorporates some 50 pieces of various sizes, ranging in thickness from 21 B.W.G. to $\frac{1}{8}$ in plate.

Since an open body of this type lacks the depth that the roof and door pillars give to a closed car, its structure must include members so designed and placed as to impart the necessary stiffness. The Sprite has a single-sheet floor, which is below propeller shaft level and is almost flat, except for shallow, sloping depressions near the front, under the occupants' feet, and for a set of four transverse depressed flutes on each side of the propeller shaft tunnel. Positioned transversely over the middle of the floor pressing, and welded to it, is a full-width cross member of top-hat section, the ends of which have outwardly turned flanges. In each end of the cross member is mounted one of the tubes that protrude through the body sill inner panels to form jacking points. Each of these panels is welded to the upturned edge of the floor pressing.

In front of the cross member, the floor pressing is cut away to clear the gearbox; the opening is flanked longitudinally on each side by a top-hat member welded to the floor as well as to the cross member. These longitudinals extend forward beyond the floor almost to the front of the car, and they carry the engine, the radiator and the spring abutment brackets. Welded under the overhanging portion of each longitudinal is a closing plate. The resulting box-section arms are bridged near their leading ends by a wide but shallow member of top-hat section, which also has a welded-on closing plate. When the power unit is installed, this cross member lies just ahead of the sump, which it partially masks, as regards cooling, but also protects to some extent from damage.

In line with the bridge member, the abutment brackets for the front springs are welded to the outer faces of the

SPECIFICATION

ENGINE: Four cylinders. Bore and stroke, 62·9 mm and 76·2 mm. Swept volume, 948 cm³. Maximum b.h.p. (gross), $42\frac{1}{2}$ at 5,250 r.p.m. Maximum b.m.e.p, 136 lb/in² at 3,300 r.p.m. Maximum torque, 52 lb-ft at 3,300 r.p.m. Compression ratio, 8·3 : 1. Three-bearing, counterbalanced crankshaft. Chain driven camshaft. Vertical, in-line overhead valves actuated by push rods and rockers. Heart-shape combustion chambers in cast iron cylinder head. Siamesed inlet ports; exhaust ports of cylinders 2 and 3 siamesed. Two S.U. H1 semi-downdraught carburettors, with $1\frac{1}{8}$ in diameter throttle barrels, mounted on aluminium manifold embodying balance pipe. Pancake type air filters. AC mechanical fuel pump. Dry weight, with flywheel and clutch, 245 lb.

TRANSMISSION: Borg and Beck single-dry-plate clutch, $6\frac{1}{4}$ in diameter, housed in cast aluminium bell housing integral with gearbox shell. Four-speed gearbox with synchromesh on top, third and second gears. Remote type manual control. Standard ratios: top, 1 : 1; third, 1·412 : 1; second, 2·374 : 1; first, 3·627 : 1; reverse, 4·66 : 1. Competition ratios: top, 1 : 1; third, 1·357 : 1; second, 1·916 : 1; first, 3·20 : 1· reverse, 4·114 : 1. Dry weight, 35 lb. Hardy Spicer open propeller shaft with needle roller bearing universal joints. Three-quarter floating rear axle with banjo type casing and hypoid bevel reduction. Ratio, 4·22 : 1.

longitudinals. They are braced by a channel-section stiffener positioned transversely above the cross member. The pivot brackets for the lower wishbones of the front suspension are welded to the front and rear lower edges of the abutment brackets. Welded to the outer faces of the longitudinals, slightly ahead of the abutment brackets are the vertical, angle-section pressings that carry the radiator assembly. At a later stage, a channel-section right-angle stay, connecting the inner panel of each front wheel arch to the appropriate radiator bracket, is bolted in position.

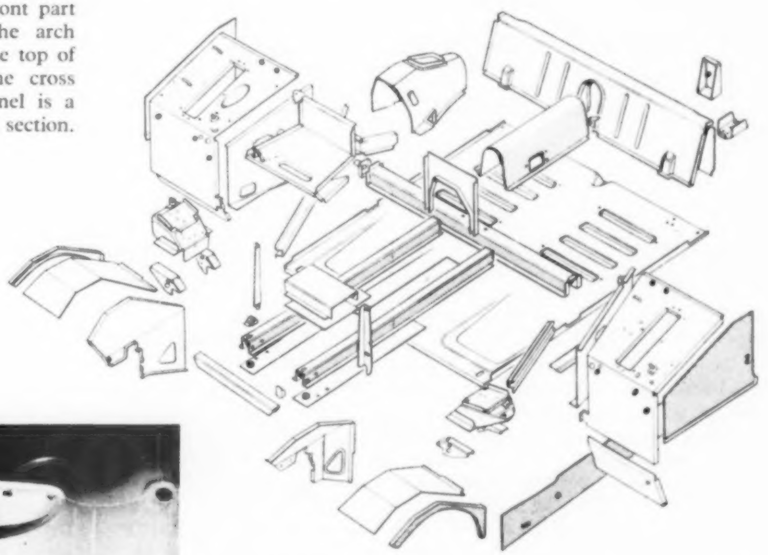
The wheel arches, which are outboard of and above the various brackets just mentioned, are each fabricated from three parts—inner, top and outer—and the rear edge of the last named is welded to the front edge of the body sill side panel. Behind the wheel arches are two assemblies that surround the occupant's legs and feet; each consists of inboard and outboard side panels and a third pressing that comprises the top and front. The front portion of this pressing forms the rear of the wheel arch. These two assemblies, together with a fabricated bridge member between them, form a rigid dash structure.

As regards the rear half of the under-frame assembly, it has already been mentioned that the propeller shaft, when installed, is above the main floor panel, which in this area is continuous from one side of the body to the other. A flanged pressing of arch shape is welded to the floor assembly just in front of the medial cross member. Behind this latter member, the two-piece transmission tunnel assembly is welded to the floor panel between the groups of flutes described previously. The front part of the tunnel, which mates with the arch pressing, is of dome shape, to clear the top of the gearbox, and seats partly on the cross member; the rear portion of the tunnel is a flanged pressing having an inverted U section.

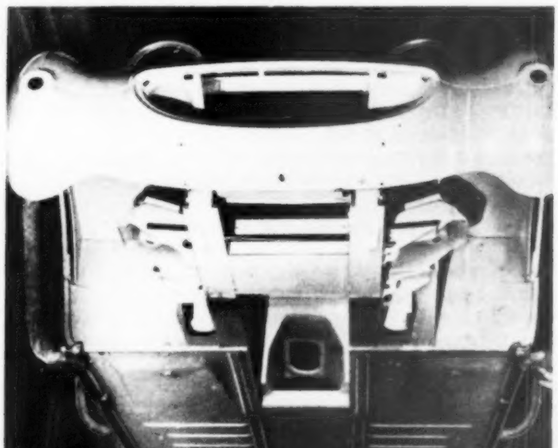
At the rear is a cross member of inverted, truncated V section, which is deeper than the rear part of the tunnel, over which it fits; its two sloping faces have flanged openings of the appropriate shape to fit round the tunnel pressing, to which they are welded. The lower edges of the cross member are flanged for welding to the floor panel. A depression is formed at the rear edge of the floor, mid-way between its sides, to provide clearance for the propeller shaft in the maximum rebound position.

When the under-frame unit is completed, it is delivered to the Swindon factory of the Pressed Steel Co. Ltd, where one of the first operations is the fitting of two mounting brackets for the rear shock absorbers. These brackets are jigged in place before being welded to the rear face of the rear cross member, one at each side. At the P.S.C. factory, the upper parts of the body are made and assembled into three units, two of which are then welded to the under-frame structure. The first of these assemblies consists of five main pressings: these are the two rear wheel arches, of 18 B.W.G. steel sheet, the floor of the boot, and two panels which eventually join the sides of this floor to the rear quarters, behind the wheel arches.

Stiffeners are fitted between the wheel arches and the floor to brace the structure. Originally, each joint was stiffened by a pair of small pressings, but these have now given place to a single larger pressing. The present support may be described as a flanged angle or W section, curved along its length to follow the contours of the floor and of the wheel arch: one flange is welded to the floor and the other to the arch.



Right: Some 50 components are embodied in the under-frame structure; the floor panel extends underneath the propeller shaft. The shaft passes through the rear cross member, which is a pressing of inverted V section. Below: In this view, the two longitudinal members, the front suspension brackets and the fluted floor are visible



The second assembly includes most of the remainder of the bodywork to the rear of the scuttle. At the back are the rear quarter pressings, between which is the rear panel; three plastics mouldings seal the joints on each side. The vertical post assembly on to which the door closes is welded to the front edge of each wing. Under the door and under the front foot of each rear wing pressing is the 18 B.W.G. outer sill panel, which extends forward to the front wheel opening. At a later stage, it is welded to the front wheel arch.

Six main parts comprise the scuttle structure that bridges the sills. On each side, the door hinge pillar consists of an inner post assembly and an outer shroud panel. At the top, the two posts are joined by the scuttle itself, which has two

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To complete the body shell, the two door assemblies are hung on their hinges and the bonnet assembly is attached to the dash by two hinge pins. The bonnet group is the its own bonnet assembly is properly attached, except that a normal steel nut is fitted to each hinge pin instead of the stiff-nut used in the finished car. For painting, the bonnet is separated from the body by removing the hinge pins, each bonnet being first marked to identify it with the body to which it belongs. When the two come together again after finishing, the stiff-nuts are fitted to the hinge pins.

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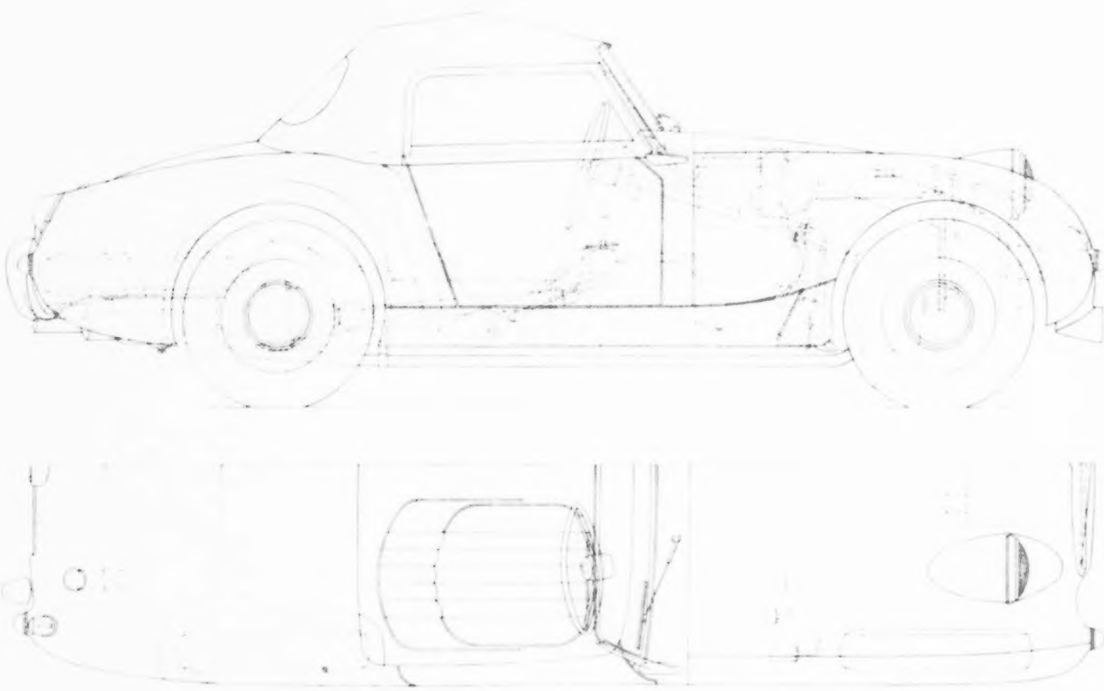

Reproduction of the general arrangement drawing of the Austin-Healey Sprite car. The wheelbase is 6 ft 8 in and the kerb weight 1,466 lb

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Although the bonnet lifts sufficiently to give good access to the various components housed underneath, it is arranged for ready detachment when necessary. This involves disconnecting the electrical harness by means of four double and two single snap connectors, grouped close to the right-hand hinge, removing one set-screw at the top of each telescopic strut, and then unscrewing the two groups of four set-screws that hold each hinge plate to the bonnet assembly. The apparently quicker method of removing the two hinge pins instead of eight set-screws is not practicable once the instrument panel is installed, since the hinge pins are hidden by this unit. However, it is used at one production stage, as is mentioned in the next paragraph.

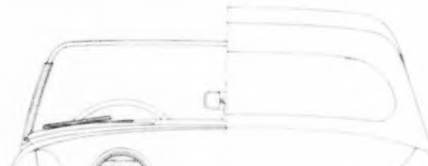
When each body leaves the Swindon works for Abingdon, panel. The upper ends of the telescopic struts are attached about half-way forward under the bonnet shoulder. Each of these struts embodies two lipped channel-section members, arranged so that one can slide inside the other; a conventional trip catch is provided. The bottom of the lower member of the strut is permanently hinged to a bracket attached to the under-frame assembly by two screws in one plane and a third at right angles, to prevent rocking. The top of the upper member of the strut is hinged by a shouldered bolt to a bracket attached to the underside of the bonnet assembly.

At its front bottom edge, the bonnet has two peg-ended bolts; these register in holes in the longitudinals of the under-frame. The bonnet-lock has two sliding latch-bolts, operated by a handle with a double-lever, which register in bushed holes in striker plates secured to the front ends of the longitudinals. With regard to the safety catch previously mentioned, this is a single pressing of bell-crank type with a hook on the end of one arm, and this hook engages a peg on the nearside wheel arch. The other arm of the catch is bent over to make a convenient operating lever.

As will probably have been gathered, the boot of this open two-seater body does not have an external lid. Access to its interior is gained from inside the body, behind the

seats; the individual squabs can be folded forward for this purpose. The spare wheel is carried in a horizontal position in the tail of the compartment, in which the hood is also stowed.

The hood is completely detachable. It is made from plastics coated fabric and is anchored to the rear edge of the cockpit by turn-buttons and press-studs. Originally, the front edge of the hood was attached to the windscreen frame by another nine press-studs, equally spaced. Even with this fairly close spacing, however, it was found that rain sometimes entered between the two components. The method



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The fuel tank comprises a shallow rectangular pressing welded under a flat panel. It is suspended directly beneath the floor by six studs fixed thereto and passing through the peripheral flange of the tank. One stud is situated near each corner of the tank, and the remaining two are in the middle of the longer, longitudinal sides. Each stud is fitted with a plain and a shakeproof washer under the nut.

Engine

The engine of the Sprite is a two-carburettor version of the well known B.M.C. Series A four-cylinder unit of 62·9 mm bore and 76·2 mm stroke, the swept volume of which is 948 cm^3, or 57·87 in^3. With its standard compression ratio of 8·3:1 it develops 42½ b.h.p. gross at 5,250 r.p.m, as against the 34 b.h.p. at 4,750 r.p.m. of the single-carburettor model fitted to the Austin A.35 and A.40 cars. Valve crash occurs at 5,900 r.p.m, at which speed the power curve is still close to its peak.

A maximum torque of approximately 52 lb-ft at 3,300 r.p.m. is quoted, and the torque curve does not fall below 50 lb-ft from 2,100 r.p.m. to 4,100 r.p.m. The corresponding b.m.e.p. figures are 136 lb/in^2 maximum and 130 lb/in^2 minimum between the speeds quoted. There is relatively little variation in the fuel consumption over the engine speed range. At 1,500 r.p.m. it is 0·538 pt/b.h.p-hr; it falls to 0·508 at 2,500 to 2,800 r.p.m. and rises again to 0·538 at 4,600 r.p.m. From the point of view of combustion efficiency, these figures need to be considered in conjunction with those for the mechanical efficiency. The curve of this efficiency shows a value of 86 per cent at 1,000 r.p.m, falling in a nearly straight line through 84 per cent at 2,400 r.p.m. and 81½ per cent at 3,000 r.p.m, then descending a little more steeply to 77½ per cent at 4,000 r.p.m. and 72½ per cent at 5,000 r.p.m.

The quoted performance figures were measured with an engine fitted with the normal pancake type air filters and a distributor with an automatic centrifugal advance mechanism giving a straight-line variation from 23 deg early at 1,500 r.p.m. to 35½ deg early at 5,000 r.p.m. As an indication of top gear road performance, the output can be compared with the tractive resistance, as shown on one of the accompanying illustrations. With six gallons of fuel in the tank, the Sprite has a kerb weight of 1,466 lb and a laden weight of about 16 cwt; its frontal area is 13·8 ft^2 and its drag coefficient is 0·00136. Tyres of 5.20-13 in section are fitted and the axle ratio is 38:9 or 4·22:1.

It can be seen from the graph that the power available and power required curves intersect at 84·4 m.p.h. The accelerative ability at lower speeds depends, of course, upon the surplus power available, as indicated by the vertical interval between the two curves. At 30 m.p.h, this figure is 13 b.h.p; at 40 m.p.h, 17 b.h.p; at 50 m.p.h, 19¼ b.h.p; at 60 m.p.h, 18 b.h.p. and at 70 m.p.h, 13¼ b.h.p. In terms of the surplus torque, these figures become 35, 34·3, 31·1, 24·2 and 15·3 lb-ft respectively at the five road speeds quoted.

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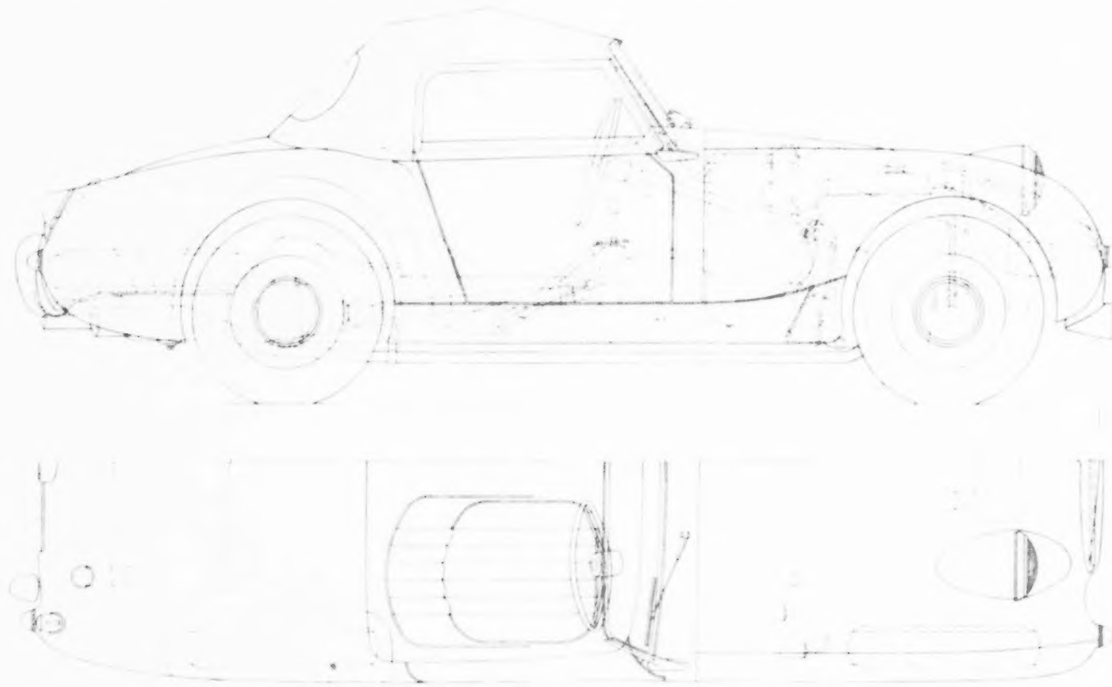

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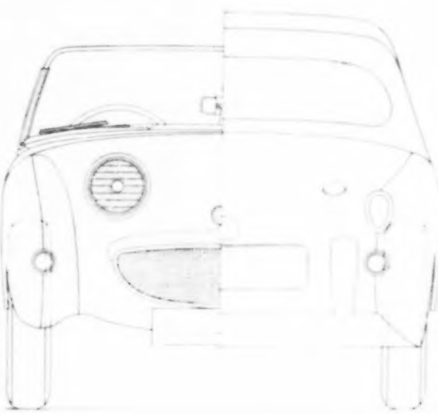

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vertical, on the other side of the chamber, between the two valves.

Particulars of the valves and springs are given in the accompanying table. The valves are actuated by push rods, and the camshaft is slightly lower than the bottom ends of the cylinders. For adjustment of the clearance there is a screw and locknut at one end of each rocker; the screw has a ball-end that seats in the cup-end of the push rod. The valve timing is checked with 0·019 in valve clearance—0·007 in more than the normal cold clearance. It is as follows: inlet opens 5 deg before t.d.c, closes 45 deg after b.d.c; exhaust opens 40 deg before b.d.c, closes 10 deg after t.d.c.

The rockers pivot on a tubular shaft carried by four pedestals, each of which is secured to the cylinder head by a pair of studs and nuts. Six of the rockers are disposed in pairs on the three portions of the shaft between adjacent brackets, those of each pair being separated by a light helical spring. The two remaining rockers are carried one on each overhanging end of the shaft, where each is retained by a double-coil spring washer, plain washer and split pin.

A single-row roller chain of $\frac{3}{8}$ in pitch drives the camshaft, and a degree of automatic tensioning is effected by the simple expedient of fitting a synthetic rubber ring in a peripheral groove machined on each side of the camshaft chain wheel. These rings have an overall installed diameter rather greater than that of a circle touching the side plates of the chain when it is on the sprocket. The resultant compression of the rings by the chain is sufficient to ensure that the latter's effective pitch circle on the wheel is increased as the pitch of the chain increases with wear.

Three bearings carry the camshaft. The front one has a Vandervell D2 white metal liner, but in the other two the shaft runs directly in the crankcase material. Axial thrust is taken at the front by a plate secured to the casting by three set-screws.

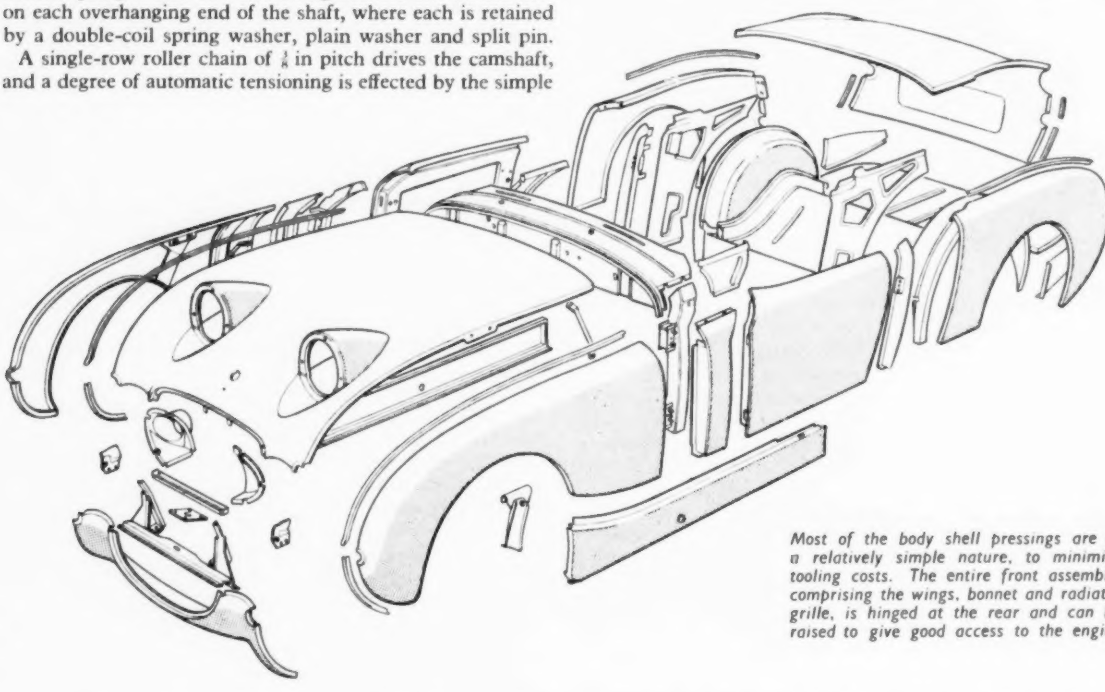
The forged, counterbalanced crankshaft is carried in three Vandervell D2 white metal bearings of 1·75 in diameter and 1·188 in length. In conformity with general B.M.C. practice, the big-end bearings, which have a diameter of 1·625 in and a length of 0·875 in, are of the VP3 copper-lead type, and the stepped cap joint is inclined at 45 deg to the longitudinal axis of the rod. Each big end is offset axially by about 0·15 in from the longitudinal axis of the rod and piston.

Axial location of the crankshaft is provided by a diametrically divided thrust washer on each side of the centre main bearing.

The $\frac{5}{8}$ in diameter gudgeon pin is clamped in the small end of the connecting rod by the usual split eye and pinch bolt arrangement. This bolt also provides positive location of the gudgeon pin, cotter fashion. The connecting rod length between centres is $5\frac{3}{4}$ in, so the ratio of that length to the stroke is 1·92 : 1.

Four rings are fitted to each of the split-skirt light alloy pistons. The uppermost of the three compression rings has a parallel rubbing face, but the other two have taper faces. All three are approximately 0·07 in wide. The bottom ring is a slotted scraper component, with a width of approximately 0·125 in.

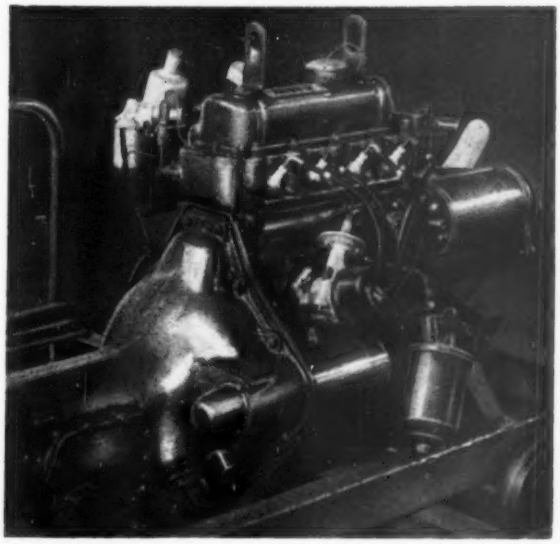
Two types of oil pump are used alternatively. One has an eccentric rotor and the other has sliding vanes. In both cases, the normal pressure is 60 lb/in^2 and the minimum pressure, for idling, is 15 lb/in^2. The pump shaft fits freely into the hollow tail end of the camshaft and the drive is transmitted by a diametral pin. This pin registers in a slot machined in the end of the pump shaft.

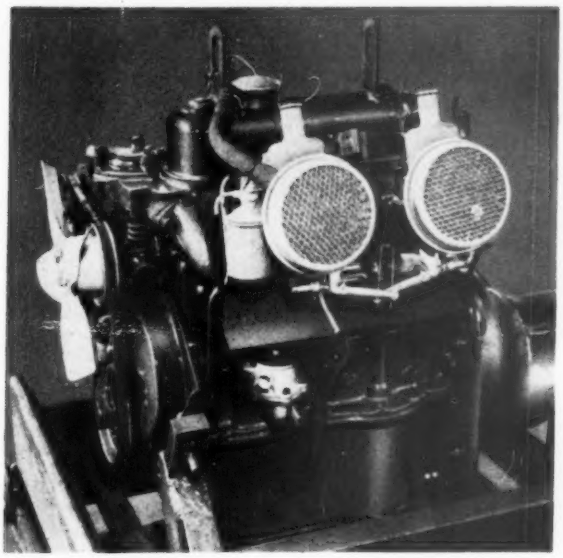


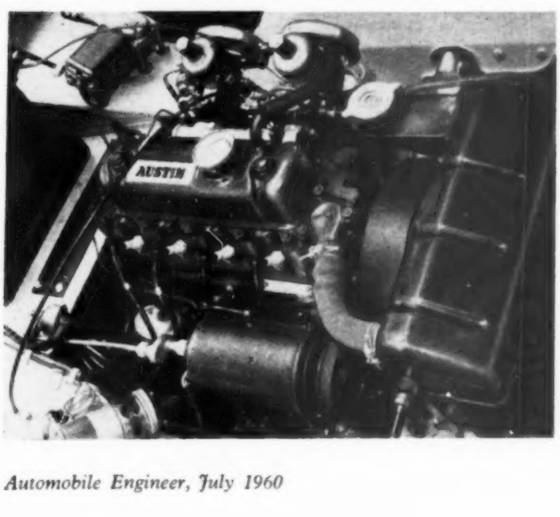
Most of the body shell pressings are of a relatively simple nature, to minimize tooling costs. The entire front assembly, comprising the wings, bonnet and radiator grille, is hinged at the rear and can be raised to give good access to the engine

Oil is drawn from a six-pint pressed steel sump through a gauze strainer. It is discharged past a non-adjustable pressure relief valve and through a Tecalemit or Purolator full-flow felt filter, the by-pass valve of which is set to open when the pressure across the filter reaches 15 to 20 lb/in^2. From the filter, oil is fed to each main bearing, to each camshaft bearing and to the front pedestal of the hollow rocker shaft, along which the lubricant flows to the drilled radial holes feeding each rocker bearing.

Two S.U. H1 semi-downdraught carburettors, with $1\frac{1}{8}$ in diameter throttle barrels, are fitted. They are inclined at 20 deg to the horizontal, on the left side of the engine. The inlet manifold is a light alloy casting consisting of two short port branches joined by a horizontal balance pipe. Each port branch connects one carburettor to a siamesed port in the cylinder head, one of which divides to serve the inlet valves of cylinders 1 and 2, and the other those of cylinders







3 and 4. The balance passage is connected by a short vertical tract to the roof of each port branch. It is of relatively large bore, about ¾ in, for most of its length but is restricted to about half that diameter mid-way between its ends. Each carburettor is fitted with a Cooper pancake type air cleaner, and the two are fed by an AC mechanical pump. A three-branch manifold connects the exhaust ports in the head to the down pipe. Cylinders 1 and 4 have separate exhaust ports, but 2 and 3 discharge into a siamesed port connected to the short central branch of the manifold.

Clutch and gearbox

Four dowel bolts secure the flywheel to the flange at the rear end of the crankshaft, which is bored out for the bush carrying the leading end of the gearbox first-motion shaft. The clutch is a standard Borg and Beck 6¼ in diameter unit with a single dry plate, on which are linings ⅛ in thick and with a total frictional area of 30·1 in^2. It is controlled hydraulically. Six springs are employed, to exert a total axial force of 540 to 600 lb, and the pressure plate travel to full release is 0·24 to 0·27 in.

The gearbox shell is integral with the clutch bell housing and is cast in B.S. 1490 LM4M or LM21M aluminium alloy. An orthodox gearbox layout is employed, in that the first- and third-motion shafts are in line, giving a direct drive in top gear. The layshaft is to the left of and slightly lower than the mainshaft, and the reverse shaft is almost directly under the mainshaft.

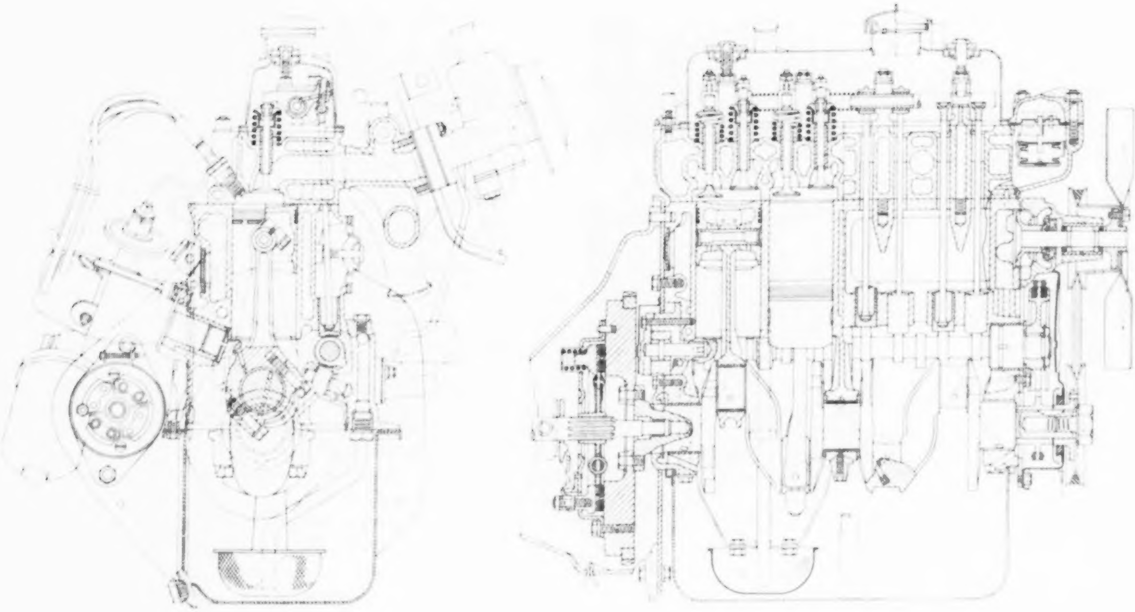
Both the layshaft and the reverse shaft are stationary, their gears revolving on them. The reverse cluster rotates on a plain bush but the layshaft cluster is carried on two needle roller bearings, each with 23 rollers 0·098 in diameter and approximately 0·78 in long; the internal diameter of their housing is approximately 0·83 in. The cluster itself consists of three single-helical gears and a straight-tooth pinion, all formed on a single forging. Details of the various gears are given in the accompanying table. For competition work, closer ratios are available.

The front end of the mainshaft is carried in a plain bearing housed in the rear end of the first-motion shaft, and its rear end is carried in a ball journal bearing. Another ball bearing locates the first-motion shaft, where it enters

Three views of the two-carburettor version of the B.M.C. 948 cm³ Series A power unit. Top: The distributor is mounted low and nearly horizontal; the clutch housing and gearbox shell are one aluminium casting. Middle: Each S.U. carburettor has a pancake type air filter. Bottom: The cooling fan is shrouded to increase its efficiency, and the radiator filler cap is sited on an extension of the header tank

VALVE DATA

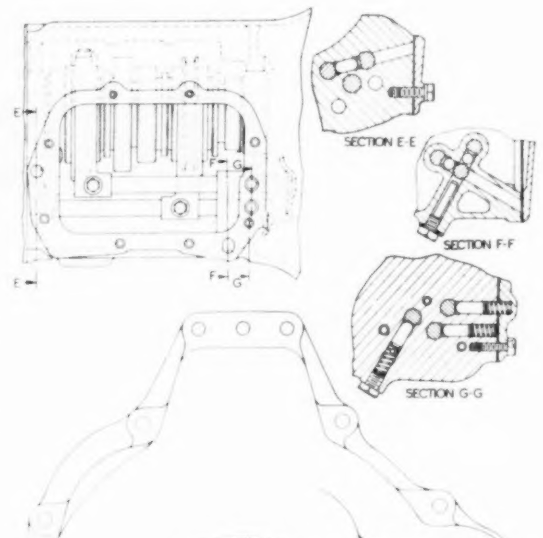
	Inlet	Exhaust
Material	En.52	21-4-NS or XB, Stellite faced
Head diameter	1·094 in	1·00 in
Throat diameter	0·969 in	0·908 in
Stem diameter	0·2793-0·2798 in	0·2788-0·2793 in
Diametral clearance in guide	0·0015-0·0025 in	0·002-0·003 in
Seat angle	45 deg	45 deg
Face width—on valve	0·055-0·075 in	0·055-0·075 in
on seat	0·036-0·056 in	0·052-0·072 in
Valve lift	0·28 in	
Spring material	En.49	
Number of working coils	4½	
Coil diameter	1·118-1·132 in	
Wire gauge	9½ S.W.G. (0·136 in)	
Spring rate	114 lb/in	
Spring length—free	1¾ in	
installed	1$\frac{19}{32}$ in	
Spring load—valve seated	52½ lb	
Valve crash speed	5,900 r.p.m.	
Valve clearance—cold	0·012 in	

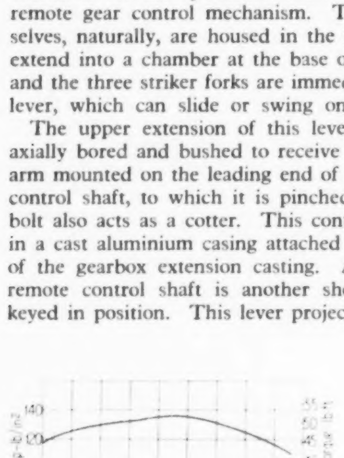


Orthodox B.M.C. practice is followed in the engine, which has a stroke : bore ratio of 1·21 : 1. The valves are vertical, heart-shape combustion chambers are employed, and the inlet ports are siamesed. Rubber rings are fitted to the camshaft sprocket to compensate for wear of the chain

the box. Assembly of this gearbox is effected through the open rear end, which is subsequently closed by a relatively long, aluminium extension casting. The third-motion shaft passes through the extension and its rear end is splined to receive the sliding coupling of the propeller shaft. In the extension are the speedometer drive gears and part of the remote gear control mechanism. The selector rods themselves, naturally, are housed in the main gearbox, but they extend into a chamber at the base of the extension casting, and the three striker forks are immediately under the striker lever, which can slide or swing on its stationary spindle.

The upper extension of this lever, above its spindle, is axially bored and bushed to receive the ball end of a short arm mounted on the leading end of the longitudinal remote control shaft, to which it is pinched and keyed; the pinch bolt also acts as a cotter. This control shaft bears directly in a cast aluminium casing attached by six studs to the top of the gearbox extension casting. At the rear end of the remote control shaft is another short lever, pinched and keyed in position. This lever projects upwards and accom-





Curves of b.h.p. torque and b.m.e.p. mechanical efficiency and specific fuel consumption for a Sprite engine fitted with its standard air filters

modates the ball-end at the bottom of the gear lever itself, which is in a spherical mounting in a short turret formed on the top of the remote control casing, at the rear. The gear lever is retained by a light pressing secured to the turret by three set-screws.

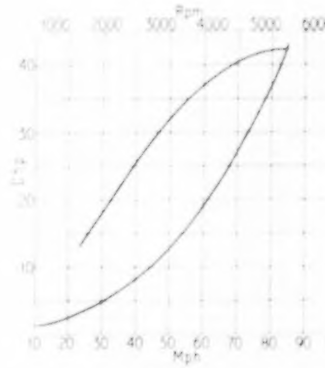
A spring-loaded plunger, housed in the rear of the turret, bears on the pivot ball of the gear lever, to prevent rattle without introducing excessive friction. Alongside it is the reverse stop, which has two main elements: on the right-hand side of the lever on the rear end of the remote control shaft is a horizontal arm, which tilts upward when the gear lever is moved towards the reverse gear position. Above the arm, and housed in the turret behind the pivot ball, is a vertical, spring-loaded plunger. When the gear lever is moved to engage a forward ratio, a spring-loaded ball registers in a notch in the plunger. The engagement of reverse, therefore, involves not merely the compression of the plunger spring but also the disengagement of the detent ball from the notch.

Propeller shaft and final drive

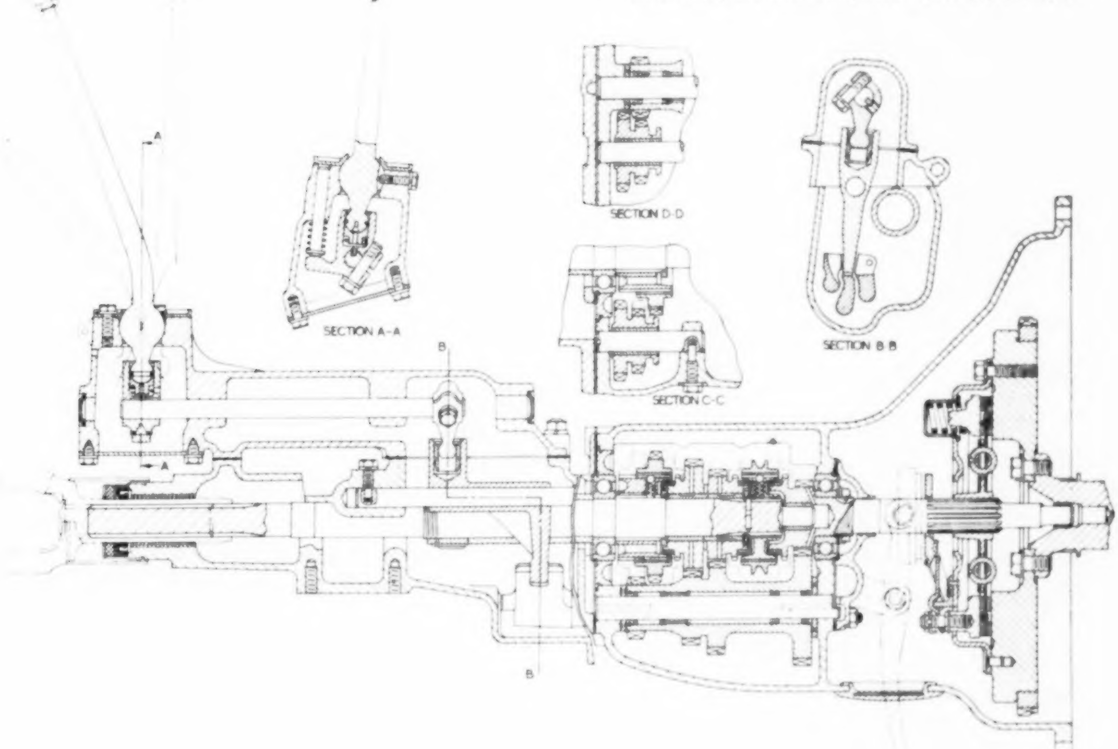
A Hardy Spicer open propeller shaft, with needle roller bearings in the universal joints, transmits the power to the rear axle, which has a banjo casing of conventional design. The final drive unit embodies hypoid bevels and a two-pinion differential. As has already been mentioned, the crown wheel and pinion have 38 and 9 teeth respectively; the planetary gears of the differential have 10 teeth and the half-shaft gears have 17 teeth. Each three-quarter floating half-shaft is integral with the flange carrying the wheel hub.

The entire bevel and differential mechanism is assembled as a unit in its own carrier, which fits into the open front of the banjo casing, where it is retained by eight studs and nuts around the banjo opening. In the gear carrier, two taper roller bearings support the hypoid bevel pinion shaft, the axis of which is offset 0·75 in below that of the crown wheel. The bores of the front and rear bearings are respectively 1·0 in and 1·125 in. Between the two inner races is a distance piece. Tightening of the flange nut on the front end of the pinion shaft clamps the assembly—comprising the two inner races, the distance piece and the universal joint flange—against a thrust washer immediately ahead of the pinion. The two outer races of the bearings are located in the carrier by shoulders.

To obtain the correct setting of the pinion relative to the



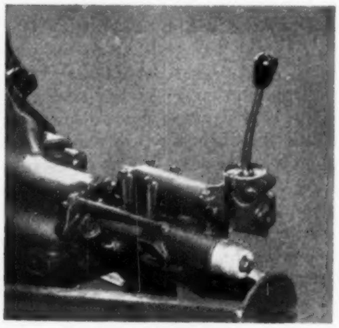
The gearing chosen is such that the curve of power required is cut by the b.h.p. curve at the latter's peak



A conventional three-shaft layout is used in the B.M.C. Series A four-speed gearbox, in which the layshaft is mounted to the left of the mainshaft and slightly below it. There is synchromesh on the upper three ratios, and alternative, closer ratios are available

GEARBOX DATA

	Number of teeth	Diametral pitch	Helix Angle	Material	Blank thickness at p.c. diameter
Mainshaft gears					
fourth	19	10	40 deg	HS 13 R	11/16 in
third	23	10	40 deg	HS 13 R	0·490 in
second	29	10	40 deg	HS 13 R	0·490 in
first	32	10	—	HS 13 R	0·369 in
Layshaft gears					
fourth	28	10	40 deg	HS 13 R	15/32 in
third	24	10	40 deg	HS 13 R	15/32 in
second	18	10	40 deg	HS 13 R	15/32 in
first	13	10	—	HS 13 R	15/32 in
Reverse idler	18 and 24	10	—	AS 24	—



A neat remote control brings the short gear lever to a convenient position. The linkage is simple and robust

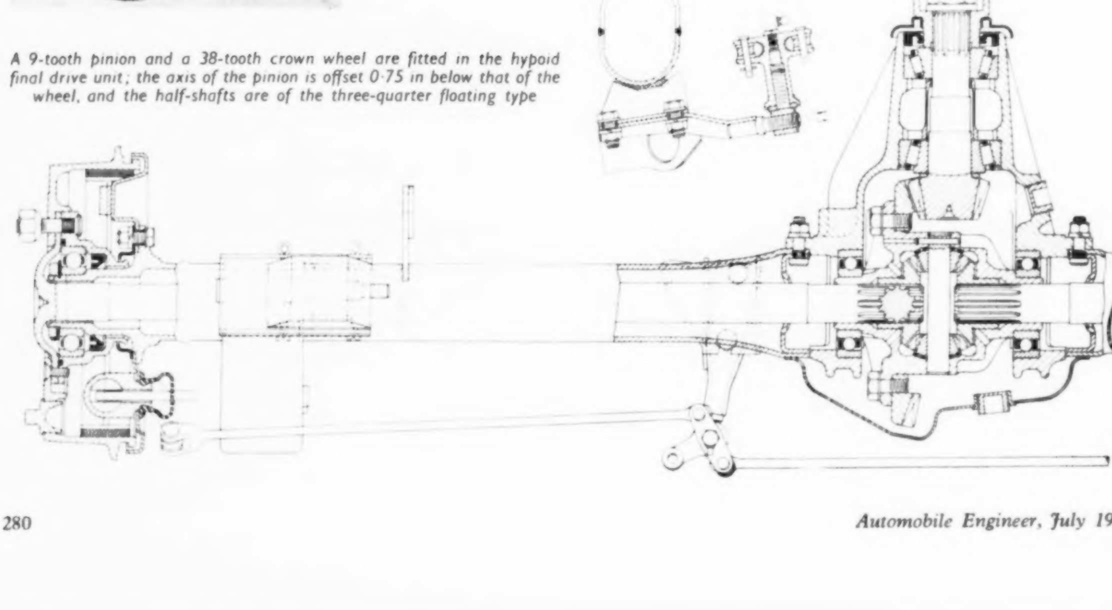
The hypoid bevels and the differential form a single unit housed in the nose casing of the rear axle. In the illustration, the gears are being checked on the bench for backlash

crown wheel, the thrust washer is manufactured in various thicknesses. Control of the pre-loading of the two bearings is obtained by inserting shims of suitable thickness between the inner race of the front bearing and the distance piece. When the flange nut is tightened to a torque wrench reading of 1,680 lb-in, a torque of 13 lb-in should be necessary to overcome the effect of the pre-load.

Two ball thrust bearings carry the crown wheel and differential cage assembly. The outer races of these bearings are secured in the carrier by caps, each retained by two studs with nuts and spring washers. These caps and the bearing seats in the carrier are shouldered so that the bearings cannot move outward, as they might do if retained solely by frictional grip. Pre-loading of these bearings can, therefore, be assured by suitable axial location of the inner races, which are carried on extensions of the differential cage. To obtain the correct location, shims are fitted between each inner race and a shoulder on the cage. The choice of shim thickness, to provide the necessary pinch of 0·002 in, is based on direct measurement of certain dimensions in each individual unit.

At each end, the axle casing carries a ball journal bearing, the inner race of which is held against an external shoulder on the casing by a nut. The end of the casing is threaded to take the nut, which is secured by a locking washer. The outer race of the bearing is clamped between the flange on the end of the half-shaft and a shoulder inside the hub, which is attached to the flange by the two screws retaining the brake drum and by the four wheel studs and nuts. A spring-loaded, moulded lip type oil seal is housed inside the hub, immediately inboard of the ball bearing.

A 9-tooth pinion and a 38-tooth crown wheel are fitted in the hypoid final drive unit; the axis of the pinion is offset 0·75 in below that of the wheel, and the half-shafts are of the three-quarter floating type



Carburation and Performance

Part II. The Zenith Range of Carburettors and How They are Matched to the Engines

CHARLES H. FISHER, M.I.Mech.E., M.S.A.E.

WHILE the Zenith Company produces an immense variety of different types and sizes of carburettor, currently at a rate of about 6,000 per day, the number of basic models employed on the popular motor vehicles is relatively small. There are, of course, variations on these basic models, but for the purpose of this article we need discuss only a few.

V type

This is a design that forms the basis for a large number of different carburettors fitted to road vehicles. One of its features is that the float chamber which contains, in addition to the float, the metering jets and most of the metering elements, is a separate component and is secured by two hexagon-headed bolts to the main body. On removal of these bolts, the float chamber comes away for ready examination and cleaning of the jets and passages, Fig. 6.

The 30-VIG range

Several millions of this design have been produced. The float system is conventional, and the fuel for the main metering system is regulated by two fixed jet orifices screwed into the floor of the float chamber, Fig. 6. These are known as the main jet and the compensator jet, and they are indicated as items 1 and 2 in Fig. 7B. Both jets feed through various channels into a common discharge orifice, and so out into the air passage of the venturi, at 3 in Fig. 7B.

Air for emulsion is admitted into the capacity well at 4, and the construction of the channels is such that the fuel metered by the compensator jet is mixed with emulsion air to a greater extent than is the fuel from the main jet. The emulsion of fuel and air issuing from the discharge beak at 3 is spread across the air passage of the venturi by a distributor bar, which is a fixture in the venturi. In certain instances, an extra distributor bar is employed: its end faces the outlet of the beak, and it is thus at right angles to the fixed bar.

The idle system follows orthodox lines, the slow running jet 5 drawing its fuel supply from downstream of the main metering system. A small bleed hole above the slow running jets admits a little air, and the resulting emulsion is passed down to the idle discharge and progression orifices at 6 and 7 respectively, in Fig. 7A. Slow running mixture adjustment may be effected either by a volume control, as shown at 8 in Fig. 7A, or by an air control screw, as at 9 in Fig. 7C.

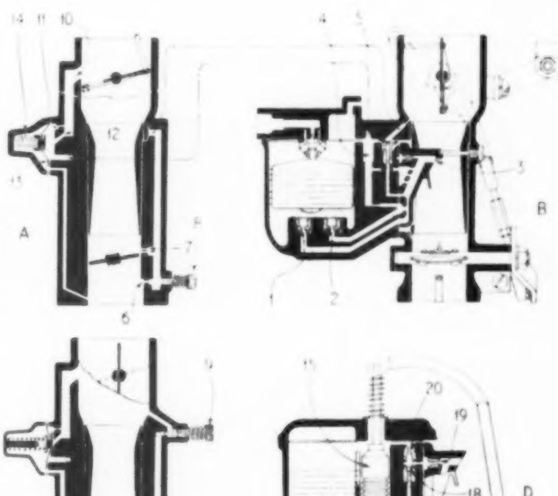


Fig. 7. Sectional diagrams showing details of the 30-VIG carburettor

Over the years, several different methods have been employed on Zenith carburettors to achieve part throttle economy. With any Zenith economizer system it is possible to match closely the part throttle requirements of the engine, no matter how lean these requirements may be: since, as has been mentioned in an earlier section, there is practically no weak limit to the mixture that the carburettor can be made to meter, the limitation on part throttle economy lies in the ability of the engine to operate on lean mixtures.

When matching the VIG series carburettors—as with other types—to an engine, it is necessary first to determine the lean limit acceptable by the engine. This is usually done by taking a series of mixture loops over the throttle range. A typical curve, taken at part throttle, is illustrated in Fig. 8. These curves are generally obtained at constant load and speed, the only variables being the mixture strength and induction system depression. The first mentioned variable is regulated by an adjustable jet and the second by the throttle.

With the VIG unit, jet adjustment is catered for by fitting a modified float chamber bowl having an adjustable needle valve screwed into its base in such a manner that the needle enters the main jet orifice. To facilitate adjustment, the lower end of the needle is knurled, and the setting can, therefore, be altered easily while the engine is running. It is usual to plot brake specific fuel consumption against induction depression to obtain the curves.

In Fig. 8, the point *X* represents the brake specific fuel

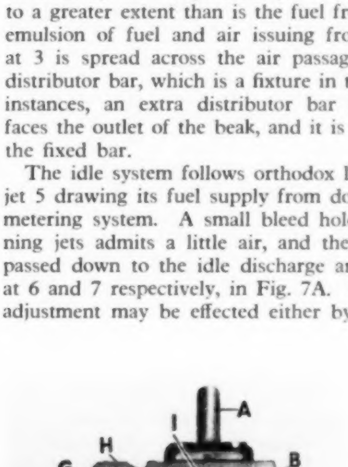
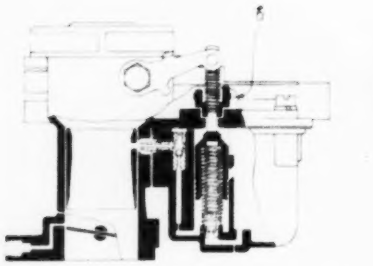
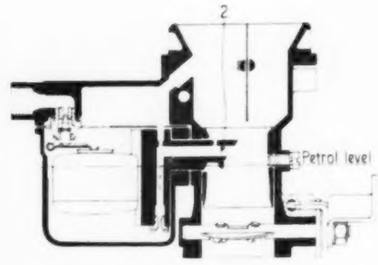
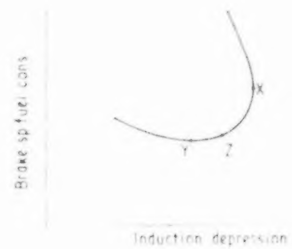


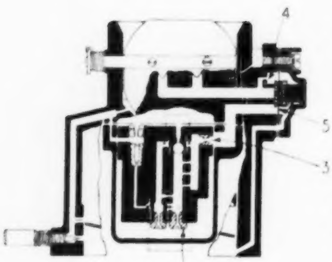
Fig. 6. Plan view of the detachable float chamber of the 30-VIG type of carburettor

A emulsion block; B capacity well; C compensating jet; D acceleration pump inlet valve; E main jet; F pump retaining screw; G acceleration pump; H pump discharge valve; I slow running jet









Right: Fig. 8. A typical mixture loop, taken at part throttle; a series of such loops is needed to match a carburettor to an engine

Above: Fig. 9. One of the newer Zenith downdraught carburettors, the VN type; a difference from the VIG series is in the position of the emulsion block

consumption for maximum power output at the throttle opening at which the curve was taken, while the point Y is that of maximum economy. This illustration clearly exposes the fallacy of attempting to gain fuel economy by running with a mixture leaner than that which the engine can usefully accept, for if the mixture is weakened beyond point Y, the fuel consumption increases — verification of this can be obtained readily by observation of the fuel consumption measuring instruments when the engine is running on a test bed. Having obtained a family of these curves, over the part load range, one has a clear picture of the absolute lean limit of mixture that the engine can use under any condition. Power units vary considerably in this respect, as between different designs; in general, loops with tails that remain fairly flat give promise of more stable carburation than those that rise sharply from the point Y, as the depression is reduced.

In selecting the fixed jets and other metering devices with which to obtain the required characteristics, it is usual to choose points on the loops, such as Z in Fig. 8, slightly richer than the lean limit at Y. This is to allow for production tolerances in respect of carburettor flow and the mechanical details of the engines. It is this final fixed-jet metering curve that produces the chain dotted, part throttle flow curve C in Fig. 5. What really determines the weak mixture limitation for part throttle running is the onset of unstable running. Sometimes the mixture strength can be weakened to the point Y and further without instability. On the other hand, some engines show pronounced instability and misfiring at a point only slightly below X, and in such cases it is not worth while to employ an economizer type of carburettor.

In Fig. 7 can be seen the method employed in the Zenith VIG model to meet these requirements. Mention has already been made of air for emulsion being inspired into the capacity well and thence into the emulsion system: the supply of air is drawn into the orifice 10 in the air intake and thence through one or both of the air ventilation holes 11 and 12, and so to the capacity well 4. A small disc valve, mounted on a flexible diaphragm 13 is spring-loaded in a suction chamber in such a manner that, when there is no depression in the chamber, the disc valve is held upon its seating by the spring 14. This action seals off the part throttle ventilation orifice 12, leaving open the smaller one 11, which is known as the full throttle bleed. Engine depression is conveyed through a channel, which can be seen clearly in the illustration, to the suction chamber.

Thus, with the throttle wide open, the suction diaphragm valve is held against its seating by the spring, and the large ventilation hole 12 is closed, so the air passing into the emulsion system is then limited by the restriction of the full throttle bleed 11. This provides for the supply of fuel and air mixture appropriate for operation at maximum power at any practicable speed. When the engine is operating under part throttle cruising conditions, however, the greater depression of the induction system is transmitted to the suction diaphragm 13, and lifts it, together with its valve, away from the seating, as shown in Fig. 7A. The orifice 12 will then be opened, permitting more air to enter the emulsion system, so weakening the mixture to meet part throttle requirements, which should be in the region of C in Fig. 5 and Z in Fig. 8. The determination of mixture requirements in the wide open throttle condition is carried out by means of an adjustable jet, as before, but in this instance the mixture loops are usually plotted in terms of b.m.e.p. or b.h.p. against brake specific fuel consumption.

All models of the VIG range incorporate an acceleration pump, which is mechanically operated and is a simple component. The brass piston 15, in Fig. 7D, is connected to the throttle spindle by a linkage, which can be seen on the right-hand side of that illustration. When the throttle is closed, the piston is raised, charging the pump chamber 16 by drawing fuel from the float chamber through the non-return disc valve 17. Then when the throttle is opened, the piston is pushed downwards, closing the valve 17 and forcing fuel past the non-return ball valve 18 and out into the airstream through the calibrated jet 19. There is a hole in the top of the ball valve assembly 20: this relieves the pump system of engine depression and so prevents fuel from being drawn from it when the pump is not in operation. The link that connects the pump actuating rod to the throttle spindle can be seen above the throttle stop screw 21 in Fig. 3D. When the link is in the upper hole in the throttle spindle lever, where it is retained by a small split pin, it is in the "long stroke" or "winter setting", and of course in the lower hole, it is in the "short stroke" or "summer setting" of the acceleration pump system.

VN series

This is a more recently designed range of short downdraught carburettors, for accommodation under the now fashionable low bonnets. In general, the system of metering used on the V type units is retained, although the arrangement of the components is slightly different. Most important of these differences is in the position of the emulsion block, Figs. 9 and 10: whereas on the VIG units the emulsion block is outside the float chamber, on the VN series it is placed inside the float chamber. The object of this latter arrangement is to have the jets and emulsion system immersed in fuel so that the assembly is kept cool, to avoid vapour formation and percolation troubles. The arrangement of the main and compensator jets is similar

to that of the VIG and other V type units, although the actual shape of the two jets is different: in the VN range, the main compensator jet orifices are formed in small threaded brass plugs with circular heads but, instead of the square recess, as in the VIG range, these have screwdriver slots. Another noteworthy divergence from long-established Zenith practice is in the relative sizes of the main and compensator jet bodies. On all previous Zenith models, that of the main jet is the smaller and that of the compensator jet is the larger of the two, but in the VN range the opposite is the case.

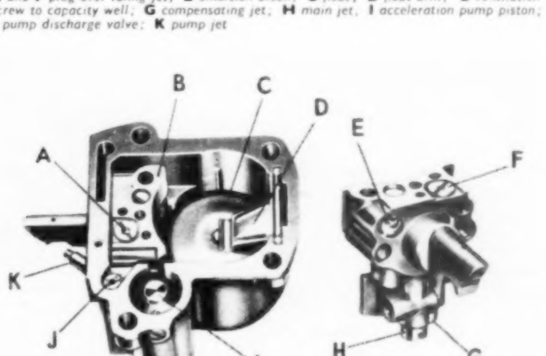
Two screws, assembled from the outside, secure the emulsion block to the inner wall of the float chamber. The right-hand illustration in Fig. 10 is of the emulsion block removed. It shows the main and compensator jets, now located at the bottom of the emulsion block, and the ventilation bleed, now at the side, for the capacity well.

From Fig. 9, it can be seen that although the disposition of the various channels is different from that of the VIG unit, the general functioning of the system is the same. The main jet 1 feeds fuel directly into the bottom of the vertical discharge tract, which leads into the horizontal beak 2. As in the VIG unit, the fuel is atomized by the entry of air through the capacity ventilation orifice 3 and into the main jet well through the three horizontal bleed holes. Also, the strengths of the full and part throttle mixtures are regulated mainly by the full throttle bleed 4 and the part throttle air bleed, which in the illustration is shown closed by the economy valve.

An acceleration pump is, of course, incorporated in the VN instruments. In essentials it is similar to the VIG

Fig. 10. The float chamber and emulsion block of the VN carburettor.

A and F plug over idling jet; B emulsion block; C float; D float arm; E ventilation screw to capacity well; G compensating jet; H main jet; I acceleration pump piston; J pump discharge valve; K pump jet



pattern, but a prolonged fuel discharge is given by the inclusion of a spring-loaded broken-beam lever. A most ingenious arrangement is employed for quickly altering the volume of fuel discharged. This is the small die-cast thimble 6, which can be lifted and rotated by the fingers, to give the pump piston a long or short stroke and so provide a summer or winter setting.

36-42 WI series

This is another recently introduced range of very short downdraught instruments. They differ from the V models in that the WI design in general follows that of the American Bendix-Stromberg instrument. This is a result of the absorption by Zenith, in 1935, of the European section of the Stromberg business.

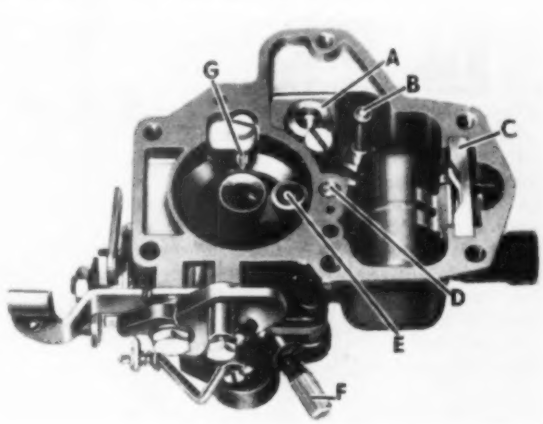
All the carburettors of this series are economy device instruments, but they differ from the V type in that the lean mixture for part throttle operation is supplied by the main metering jet, and fuel enrichment for maximum power is accomplished by means of an extra jet and fuel valve, known as the power or by-pass jet. From Fig. 12, it can be seen that the main metering jet 1 passes fuel directly into the base of the main discharge jet 2, which is also an emulsion tube. Air for emulsion is fed into the system through the main air bleed 3, the resulting emulsion being discharged into the small or primary venturi at 2.

The arrangement of the emulsion system requires some explanation, since it incorporates an anti-percolation device, designed to get rid of fuel vapour quickly. There is a division between the upper and lower portions of the discharge tube. Fuel from the main metering jet passes into the blind axial hole in the base of the tube, out through the two large radial holes into the annulus round the tube, and then through the series of small radial holes drilled from the underside of the upper portion of the tube, into the bore of that portion. The object of this construction is to keep vapour bubbles on the outside, while permitting the fuel to pass freely into the upper end of the tube; the vapour bubbles pass up and round the annulus and burst harmlessly in the domed air bleed 3. A conventional volume control type of idle system is incorporated. The general arrangement of the whole unit is shown in Fig. 11.

When carburettors of the WI series are being matched to an engine, requirements for part throttle operation are explored by plotting mixture loops, in the manner already described in connection with the V series. For this operation, the main metering jet is replaced by a screwed component that is similar to it in form but which contains a threaded adjustable needle assembly. When the mixture requirements have been determined, a size of the main metering jet is selected to fulfil those for economical cruising, and the extra fuel for high power is obtained by fitting a by-pass jet assembly of the appropriate flow characteristics. Fuel metered by the latter system flows from the by-pass jet directly into the base of the main discharge jet as shown in Fig. 12. The by-pass valve is controlled by a suction operated, spring loaded diaphragm system 4 and 5, in Fig. 12, similar to that of the VIG range. Under part throttle conditions, the engine depression holds the diaphragm stem away from the by-pass valve rod 6. As the full throttle condition is approached, the depression falls, and the by-pass valve opens, thus permitting the extra fuel to be delivered.

Fig. 11. View from above of one of the WI downdraught carburettors. The design resembles that of the U.S.A. Bendix-Stromberg instruments

A acceleration pump piston; B power jet; C float bracket; D idling jet; E high-speed bleed; F volume control screw; G pump discharge nozzle



On the WI series, the acceleration pump system is simple and calls for little comment. Fuel is discharged from the pump, into the airstream, from the small die-cast nozzle, which can be seen in Fig. 11. As in the VIG and VN pump systems, there is an air bleed over the nozzle, so that it is not subject to engine depression. On some 36-WI installations, where the pump discharge is required in the early stages of throttle movement, a cam operating linkage is employed; in others, a simple lever type linkage is used. The larger WI carburettors, the 42-WI series, are almost identical with the 36 range, except in that the acceleration pump is of the Zenith-Stromberg D.I.V. pattern, in which a follow-up device is incorporated.

Cold starting

Starting from cold demands the richest mixture of the whole operating range—the air : fuel ratio required for starting at ambient temperatures of 0 deg F is in the region of 1 : 1, by weight. When the engine has fired, sufficient air must be admitted into the engine to permit it to overcome the friction due to the high drag and the cold lubricating oil. With the strangler type of starting system, this is arranged by interconnecting the strangler and the throttle spindle so that the throttle valve is opened a certain amount when the strangler is in operation. After the engine has been running a few moments, the mixture needs to be adjusted slightly to avoid over enrichment and stalling. On the current Zenith carburettors, this is done automatically by a spring-loaded eccentrically-mounted strangler valve. In the past, however, a few carburettors were produced with an auxiliary starting carburettor but, as cold starting requirements have become more severe, the semi-automatic strangler system has proved superior. The cold starting specification may vary between one motor-car manufacturer and another, but in the majority of instances it will be found to be between 0 deg F and −20 deg F.

In the Zenith semi-automatic system, the strangler valve is pivoted eccentrically in the air passage, Fig. 13. When the strangler control is moved to the cold starting position, the strangler valve is closed by a light spring which, in the illustration, is on the right-hand end of the spindle—the only positive control that the strangler lever has over the valve is in opening it. At the same time a linkage automatically opens the throttle valve to a pre-determined position, which has been found, generally by experiment in a cold room, to be best for the cold starting condition specified. When the engine fires and runs, the ingoing airflow acting on the eccentrically pivoted strangler valve forces it open slightly against the tension of the spring, so that more air is admitted to weaken the mixture automatically. As the vehicle is driven away from cold, the increasing airflow opens the valve still further, and so automatically regulates the mixture. As the manual control is returned to the normal operating position, a tag on the mechanism forces the strangler open; at the same time, the fast idle linkage automatically reduces the throttle opening to the normal idling position.

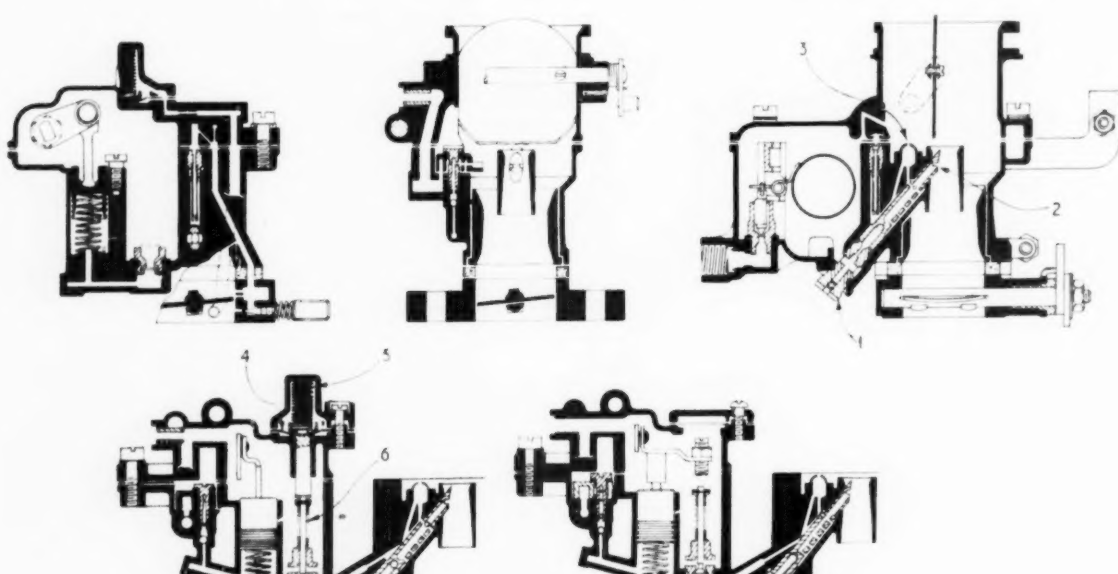
Automatic choke

The automatic choke is an ingenious device to provide the rich mixture necessary for starting the engine from cold and, also, to weaken the mixture in proportion to the rising temperature of the engine after it has started; furthermore, it sets the throttle for the cold start, and appropriately reduces the fast-idle speed as the engine temperature rises. In short, the only action the driver need take, initially, apart from switching on and operating the electric starter device, is to press the accelerator pedal once: this permits the whole assembly to snap into the cold start position.

From Fig. 14, it can be seen that the essential basic feature of this system is an eccentrically mounted strangler valve, as in the manually operated system. A coiled thermostat spring is used, instead of the simple spring of the manual system, to hold the strangler valve lightly closed when the engine is cold. As the engine warms up, the tension in the thermostat spring is automatically decreased, so permitting the strangler valve to open.

There are several arrangements of the thermostat as applied to the strangler valve; in some instances, the thermostat spring is mounted on the exhaust manifold, and in

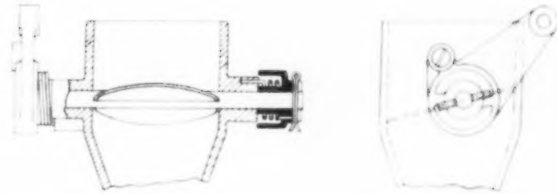
Fig. 12. Diagrammatic sectional views of the WI type carburettor. Fuel enrichment for maximum power is effected by means of a by-pass jet



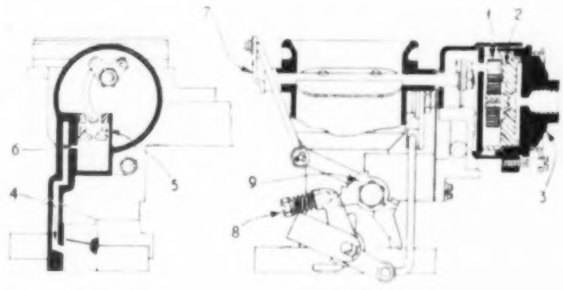
others on the water jacket round the induction system. In the system illustrated in Fig. 14, the thermostat spring 1, which is a bimetal coil, is housed in a chamber mounted on the side of the carburettor. Its inner end is anchored to the metal cup 2, and its outer end has a bent tag which is arranged relative to the pin on the strangler lever in such a manner that, when cold, the spring will hold the strangler valve closed.

The channel 4 allows the depression in the induction system to be communicated to the thermostat chamber. This depression causes hot air to be drawn from an area adjacent to the exhaust manifold, through a small bore pipe connected to the union screwed into the heat resistant retainer 3, over the thermostat spring and into the engine. To prevent over-enrichment after the cold start, a vacuum operated piston 5 is linked to the strangler spindle. When the engine fires, the rising depression in the induction system is transmitted through the channel 4 to the cylinder in which the piston operates. This draws the piston down as far as the radial hole 6, and opens the strangler valve a predetermined amount.

Linked to the strangler spindle lever, at 7, is the fast idle cam and stop-screw assembly 8 and 9. This automatically opens the throttle the appropriate amount for the cold start and subsequently for warming up. To reduce the speed progressively as the engine warms up, the cam is stepped. For adjusting the tension of the thermostat spring, the retainer 3 can be rotated, after its three clamping screws have been loosened. There is a number of serrations upon the circumference of the retainer, and a die-cast projection on the top of the thermostat housing forms a pointer which, in conjunction with the serrations, indicates the setting. An interesting feature of the assembly is the metal cup 2, which retains the heat when the engine is stopped.



Above: Fig. 13. The Zenith semi-automatic strangler system embodies an offset pivot axis. Below: Fig. 14. In the VNT automatic choke layout, the strangler valve again has an offset pivot; its position is controlled by a thermostat spring, which is in the form of a bimetal coil



Conclusion

What of the future? Which types of fuel system are likely to be employed in the years to come? Is petrol injection still to be the panacea for all fuel system problems, as it was believed by some to be only a few years ago?

In order to obtain a clearer picture of the possibilities we should first examine the American scene. Until fairly recently there was in the States the horse-power race, and the widely used V-eight engines have been developed to give, according to their manufacturers, in some instances power outputs in excess of 300 b.h.p. Very high compression ratios have been adopted, and complicated four-barrel compound carburettors are used. So long as these expensive compound carburettors are employed on American cars, there seems to be some reasonable possibility that petrol injection could in some instances be competitive despite the inherent high cost of these systems. But the American picture is changing: according to an article entitled "1960 Passenger Car Engineering Trends", by Walter G. Patton, in the *S.A.E. Journal*, December 1959, the horse-power race is over; this year, several American concerns are offering optional economy power units, and these have compression ratios of the order of 8 or 9 : 1, being designed to run on American regular grade fuels. The four-barrel carburettors are replaced by twin-barrel instruments with lean settings, and in certain instances the size of the inlet valves has been reduced.

This trend is likely to increase still further the price gap between carburettors and petrol injection. There are also other factors. Some three years ago, Chevrolet offered the Rochester continuous petrol injection system as an optional extra on their Bel-Air model. However, no more has been heard of it since. Also, Chrysler and Pontiac offered petrol injection as an optional extra, but this has been discontinued.

In Europe, Mercedes-Benz pioneered petrol injection on their 300 SL six cylinder engine, using a Bosch jerk-pump to inject fuel directly into each combustion chamber. This arrangement does at least ensure equal fuel distribution to each cylinder, although the problem of ensuring equal distribution of the air remains. However, it is, of course, extremely expensive, and there are problems associated with maintaining the necessary very close production tolerances on the fit of the plungers in their cylinders, not to mention lubrication.

It may have been these difficulties, possibly combined with others, which have led Mercedes-Benz to alter the system completely. On the 220SE, six cylinder engine, the injection pump has two plungers only. With this arrangement, one plunger supplies the front three cylinders and the other the rear three, the petrol being injected into the induction port, close to the inlet valve. The two-plunger pump runs at half crankshaft speed, so that the discharge into the tract serving the three cylinders occurs irrespective of what part of the cycle two of the three cylinders are operating on at the moment. Therefore, in effect, the action is similar to that of a continuous injection system.

In addition to the injection pump itself, there are, of course, the usual metering elements necessary to regulate the fuel supply, according to the engine requirements at any moment. All these elements have to be produced to a high degree of accuracy, and are therefore expensive. The petrol injection system on the Mercedes-Benz 220SE costs the customer in Germany about £144 more than the carburettor installation.

One of the most ingenious petrol injection systems is that of Lucas. This unit, which has been used on Jaguar racing cars, operates on the port injection principle, as does the S.U. swash-plate type system. There remains, however, the question of price. The idea of employing petrol injection as a means of supplying fuel to the engine is very old indeed: several of the earliest motor-driven vehicles were fitted with primitive petrol injection systems. In 1919, when the author was privately toying with a scheme of a rotary internal combustion engine not unlike the Wankel unit now being sponsored by N.S.U. and Curtiss-Wright, he designed a simple petrol injection system to feed fuel into the combustion space, because this appeared the only practicable method.

It may well be that we, in this country, will concentrate

our efforts on obtaining rapid acceleration characteristics up to, say, 80 m.p.h, together with improved reliability and better fuel consumption. Straws in the wind indicate that maximum speed limits may become more widespread, and designers should consider the implications. But whatever the future, surely the aim should be at letting the difference in performance as between the saloon and the sports car become wider: the complication and expense of obtaining sports car performance with the quantity produced saloon car seems to be wasteful and economically unsound.

Zenith's Fifty Years

IN 1910, a small factory was opened in Camden Town, London, for the manufacture of a new type of carburettor. It was known as the Zenith instrument, was designed and patented in France, and was noteworthy for a novel principle of mixture compensation. Tests by vehicle manufacturers proved that the Zenith instrument offered certain advantages over other types, and the first production order was placed by Albion Motors Ltd. During the company's first year, the output of carburettors was about 25 a week, a figure that has since multiplied a thousandfold.

Production increased rapidly during the next few years, and a limited company was formed in 1914, when the factory was moved to larger premises nearer the centre of London. During World War I, a Zenith carburettor was fitted to the first tank, and subsequently as standard equipment to every British tank. Another war-time development was an aero-engine carburettor.

As a result of the post-war expansion of the motor industry, additional premises were acquired near the second factory. By 1927, demand had again outrun production capacity, so this extra factory was demolished and a new three-storey building erected in its place. The industrial depression followed, and with it came a marked reduction in output. Nevertheless, by careful arrangement of the working hours, the company was able to retain all its employees.

During this lean period, development work went on, and one of its fruits was the evolution of the first Zenith carburettor to have the body pressure die cast in zinc-base alloy instead of gunmetal. This carburettor, which was known as the Type 22 FZB instrument, also featured a detachable section carrying the jets, and it became standard equipment on the Austin 7. Soon afterwards came the U type instrument and then, in 1931, the V type. The second of these marked a considerable step forward in design, since it incorporated an entirely new method of atomizing the emulsion emerging into the throttle bore. As a measure of the merit of this carburettor, it is noteworthy that its basic principles are still employed in the instruments fitted as standard equipment to a number of cars and commercial vehicles in quantity production today.

In 1935, the Zenith company took over the manufacture in Great Britain of the American designed Stromberg carburettor. It had already become evident that the two London factories were at the limit of their capacity, and the present factory near Edgware, Middlesex, was completed during that year. The other premises were then relinquished. After the move to the new home, production continued to soar, and the familiar 30 VIG carburettor was introduced in 1937; since then about three million of this type have been sold. An important development during the late 1930s was the introduction of a system of service stations throughout the country.

During World War II, the factory was, of course, very fully employed in the manufacture of carburettors for military and other purposes. A seven-day week was worked and many machines were running continuously for 24 hours a day. In 1941, a large bomb fell in front of the factory and caused quite a lot of damage, though without interrupting production for more than two hours.

One highlight of the second post-war period was the first occasion on which carburettors were operated at the South Pole: Zenith type 28G carburettors were fitted to Sir Edmund Hillary's Sno-Cat vehicles, which reached the Pole in 1958. That year, history was made at home also, in that Zenith was the first British company to make a million carburettors in 12 months. The field of activities has recently been extended with the introduction of the MX range of carburettors for smaller engines. These instruments are already fitted to several motor-cycle, scooter, outboard and agricultural engines, and have proved efficient and economical.

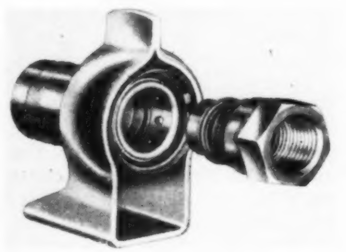
Evolution of the Car Chassis

IN THE article entitled "Evolution of the Car Chassis", which appeared in the June issue of *Automobile Engineer*, there were two errors in respect of figures. One was on the first line of the last paragraph where the date 1940 was given, whereas, of course, it ought to have been 1904, and the other was in the caption for the illustration on the opposite page, which referred to a Humber 6 h.p., instead of 16 h.p., chassis.

Quick-Release Hydraulic Coupling

AN ADDITION to the range of Aeroquip flexible hose and end fittings has been announced by Super Oil Seals and Gaskets Ltd., Kings Norton, Birmingham, 30. It is known as the Golden Flow quick-release, self-sealing coupling, and is intended for use in hydraulic feed lines having a working pressure of up to 3,000 lb/in^2. The coupling is supplied with British or American standard threads and, by the use of adaptors, is available in $\frac{3}{8}$ in and $\frac{1}{2}$ in sizes.

Each half of the coupling embodies a spring-loaded sealing valve of poppet type, and a thimble ring, also spring-loaded, for quick attachment. In the coupled position, a rubber washer effects the hydraulic seal between the two halves, which are located by a ring of hardened steel balls mounted in tapered holes. The coupling is intended both for industrial and agricultural equipment. In the second of these two applications, one half of the coupling is held by a bracket fixed to the tractor or towing vehicle, while the other is free to disengage automatically should an excessive pull be applied to the hose. The manufacturers state that tests have revealed the restriction of fluid flow through the coupling to be of negligible proportions.



This view of the new Aeroquip self-sealing coupling reveals the ring of locating balls

Tetramethyl Lead

An Additive for Improving Research Octane Numbers, especially of the Low Boiling Point Fractions of Fuels

IN a paper presented to the Summer Meeting of the Society of Automotive Engineers in Chicago, three members of the Research Department of Socony Mobil Oil Company have given the full story of their discovery of certain special properties of tetramethyl lead, TML, the new additive for the improvement of anti-knock qualities of modern petrols. Brief details have previously been given in the February 1960 issue of ***Automobile Engineer***. The value of this additive, which boosts the octane number of the lighter components of petrol, was discovered as a result of extensive work in the Company's Paulsboro, New Jersey, Research Laboratories, and road tests of many makes of cars, carried out by Mobil affiliates in England, America, France, Germany and Italy.

Tetramethyl lead is deemed to be an important addition to chemicals available to refiners for the production petrols having high road octane numbers. The authors predict that within five years it will be widely used to overcome the problem of octane depreciation of petrols, produced by the most modern refining techniques, when used in modern cars with manually controlled transmissions, and in general to increase the anti-knock quality of petrols still further.

In their paper, the authors, R. H. Perry, Jnr., C. J. DiPerna and D. J. Heath, explained that refinery methods have changed during the past 10 years: the proportion of petrol produced by catalytic cracking and catalytic reforming techniques has increased at the expense of that produced by straight-run and thermal techniques. The platinum catalyst reforming technique is the most significant recent development. Widespread use of this process has made possible most of the increase that has occurred in the past 10 years in respect of motor gasoline octane numbers. However, efficient use of the very high quality gasoline components produced by this process has presented the one formidable problem already referred to, namely road octane number depreciation. In some cars, especially those with manually controlled transmissions, gasolines containing large percentages of catalytic reformate have been found to have much lower road octane numbers than were predicted from the Research and the Motor Method ratings. A combination of two factors was found to be responsible for this; they are:

(1) Uneven anti-knock quality distribution over the whole boiling range of catalytically reformed petrol, such that a significant proportion of the petrol has a much lower anti-knock value than the overall figure. That portion of the petrol which boils at about 200 deg F sometimes has 20 octane numbers lower value than the heavier and lighter fractions.

(2) Uneven mixture distribution to the individual cylinders, through the intake manifold system. During the initial period of an acceleration from a low engine speed, there is a separation of light and heavy components in the manifold and all cylinders do not receive the same mixture so, if the anti-knock quality is not distributed uniformly among the different constituents of the petrol, some cylinders receive a mixture with a low anti-knock quality. This condition is further complicated by the fact that tetraethyl lead, TEL, because of its high boiling point, of 392 deg F, is concentrated in the less volatile components, which have high anti-knock values, at the expense of the lighter components.

It has been found that road octane number depreciation in cars with manual transmissions can be related to the following factor: (volumetric fraction of the cut from the initial boiling point to 220 deg F) × (Research octane number of the petrol) − (Research octane number of the cut from the initial boiling point to 220 deg F, but with ½ TEL content, in terms of cm^3/gal, of the petrol).

From the form of this depreciation factor, it is evident that depreciation could be reduced by either increasing the Research octane number of the lower boiling fractions or reducing the fraction of the petrol boiling below 220 deg F. Large reductions cannot be effected in the latter fraction, because this would, of course, markedly affect other performance characteristics, such as warm-up; so, after considering many ways, including the changing of refinery processing plans, it was decided that the first of these alternatives should be adopted and that this should be effected by increasing the anti-knock additive concentration in the fractions with boiling points up to 220 deg F.

In all tests carried out with TML, TEL and many intermediate lead alkyls, TML was by far the most effective in reducing the road octane number depreciation of reformate petrols. The effectiveness of the other ethyl methyl lead alkyls generally decreased in order of increasing boiling point, and TEL was the least effective. It follows that TML is most effective because it has the lowest boiling point and, in the induction manifold, is distributed with the lower octane fractions of the petrol. Iron pentacarbonyl was also very effective but was ruled out because of its adverse effects on engine wear.

In the course of the work, another important discovery was made with regard to tetramethyl lead usage: not only was TML more effective than TEL, as measured by road ratings of reformate fuels in cars with manual transmissions, but it also was a more effective anti-knock additive than TEL, as measured by Motor and the Research Method ratings. Furthermore, the advantage for TML was found to increase as the anti-knock quality of the catalytic reformate increased.

Thus, TML represents an important addition to materials available to refiners for the production of petrols having high road octane numbers. It not only possesses an advantage over TEL for increasing Research and Motor octane numbers in catalytic reformate blends, but it also enables refiners to overcome the problem of road octane depreciation peculiar to the petrols refined by catalytic reforming methods. Thus, it opens the way to the wider use of these latest and more advanced techniques for the production of hydrocarbon fuels.

Valve Life

A BOOKLET on the subject of valve life has recently been produced by the technical division of Farnborough Engineering Co. Ltd. It is well illustrated, with pictures of examples of typical valve failures, and the technical information obviously has been carefully prepared. This booklet, which can be obtained free of charge from Farnborough Engineering Co. Ltd., Farnborough, Kent, should be of interest to design, development and service engineers.

Thermal Aspects of Vehicle Braking

Heat Generated, Calculation of Temperature and of Braking Areas Required and Correlation Between Theory and Practice

T. P. NEWCOMB, M.Sc, A.Inst.P*

In the first part of this paper, a discussion is given of the heat generated during braking and the proportions absorbed by the drum and the linings.

Expressions to permit calculation of the temperature rise in a drum or disc are given for single brake applications. Typical transient temperature curves are shown for single applications with small saloon cars, high powered cars and public service vehicles. A comparison is made between drum and disc brakes, and also between cast iron drums and bimetallic drums.

In the second part, equations are given for the temperature rise during regular repeated brake applications when cooling is taken into account. It is shown how these expressions enable the area of the braking path to be calculated, for a specified maximum temperature. Consideration is given to actual brake usage measurements when vehicles are operated on road circuits, and it is shown how random braking can be reduced to equivalent cyclic applications. From one such circuit on the Alps a fade test and braking area for a sports car is derived. This area is compared with that obtained with the conventional brake rating formula.

ONE of the fundamental problems in the braking of vehicles is the determination of the temperatures reached, at the surface of contact and within the brake drum and lining, during brake applications. If temperatures become too high, a deterioration in brake performance may result from a number of causes, the chief one being that of brake fade when the coefficient of friction between lining and drum is seriously reduced. Linings may wear at a much greater rate at higher temperatures, and high thermal stresses and localized hardening of metal near the friction surface may cause surface cracking, or, in extreme cases, complete drum failure.

Some of the factors that affect the temperatures attained are the rates of deceleration and duration of braking; the road speeds at each brake application; the dimensions and physical properties of the drums and linings; the frequency of the brake applications; and the degree of air cooling from finning and wheels. However, these factors each play a different part in the various kinds of braking likely to be

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NOTATION

A, A_1, A_2, A'	*Area of outer surface of drum, rubbing path, friction material and overall area of hub and drum respectively*
a	*Deceleration*
b	*Parameter* $= Ah/WC$
C_1, C_2, C'	*Specific heat of drum material, lining material, hub and wheel material respectively*
D	*External drum diameter*
d_1	*Drum thickness*
E_1, E'_1	*Thermal equivalent of kinetic energy, of vehicle, entering unit area of front and rear drum respectively*
E_2, E'_2	*Thermal equivalent of kinetic energy, of vehicle, entering unit area of front and rear linings respectively*
E_1^*	*Thermal equivalent of kinetic energy, of vehicle, entering unit area of front disc*
f	*Compressive stress at the rubbing path at any instant during braking*
f_o	$\frac{E_1 Yl}{T(1-v)}$
g	*Acceleration due to gravity*
h, h_2, h'	*Coefficient of heat transfer of surface of drum, surface of lining plus shoe, and of the overall from the hub and drum respectively*
J	*Mechanical equivalent of heat*
K	*Overall coefficient of heat exchange*
K_1, K_2	*Thermal conductivity of drum and lining material respectively*
K'	*Constant*
l	*Coefficient of linear expansion*
M	$1/T$
N_1, N'_1, N_2, N'_2	*Thermal equivalent of energy per unit area per unit time entering a front drum* (E_1/T), *front linings* (E'_1/T), *a rear drum* (E_2/T) *and rear linings* (E'_2/T) *respectively when braking to rest*
N_1^*, N_2^*	*Heat inflow per unit area per unit time entering front and rear drums respectively when continuously braking down a uniform gradient*
n	*Number of brake applications*
P	*Weight of vehicle*
T	*Duration of brake application when vehicle comes to rest*
t	*Time*
t_o	*Regular interval between repeated brake applications*
U	*Final linear speed of vehicle after braking for* t_1 *sec*
u	*Initial linear speed of vehicle*
V	*Velocity of air*
V_m	*Mean journey speed*
v	*Linear speed at any instant*
W	*Weight of drum*
W_m	*Total work done*
W'	*Weight of hub and wheel*
$W_s C_s$	*Thermal capacity of shoe plus lining*
Y	*Modulus of elasticity of drum material*
y	*Proportion of the total braking force transmitted to the front wheel axle*
α_1	*Thermal diffusivity of drum* $= \frac{K_1}{\rho_1 C_1}$
$\Delta\theta, \Delta\theta'$	*Mean temperature increase during a brake application, drum, lining plus shoe*
θ	*Temperature rise at the rubbing path at any instant during a brake application*
θ_e	*Temperature rise at the exterior drum surface during a brake application*
θ_o	*Ambient temperature*
θ_s	*Maximum friction surface temperature during repeated braking*
θ_{av}	*Average temperature of drum at any instant*
θ_{max}	*Maximum temperature reached at the rubbing path during a brake application*
θ'', θ''_n	*Temperature at end of a braking period and value after n repeated applications at regular intervals respectively*
θ''_{limit}	*Steady state value of* θ''
v	*Poisson's ratio*
ρ_1, ρ_2	*Density: drum material, lining material*
σ	*Proportion, of total heat generated, which enters a drum*
ϕ	*Angle of inclination of hill*

encountered, that is, continuous brake applications to bring the vehicle to rest and intermittent braking to control vehicle speed in traffic and at bends or when travelling down hill. Consequently, the temperatures reached in each kind of braking must be determined.

This can be done by calculating the amount of energy dissipated in the form of heat at the interface of drum and lining and determining how the energy is shared between the contacting bodies. The problem then reduces to one of heat conduction through a solid, in the shape of a hollow cylinder—which can be developed into a slab—and its solution permits calculation of the temperature reached during an individual brake application. A knowledge of these temperatures in drum and lining, together with cooling losses from these bodies, can then be utilized to derive expressions for the temperature rise in repeated braking at regular intervals. The temperatures occurring in disc brakes can be derived in a similar manner by considering the heat flow through a slab.

During intermittent braking, from a given initial condition of vehicle energy, the temperature attained depends mainly on the coefficient of heat transfer from the drum to the atmosphere and on the area of rubbing path inside the drum. If the heat transfer coefficient is known, or can be estimated, then by imposing a limiting temperature that it is desirable for the friction surface not to exceed, the area of braking path can be determined. Before this can be done, patterns of driving behaviour over a journey must be considered and, in particular, the amount of energy likely to be dissipated in braking under the most severe conditions possible estimated. Thus random braking must be reduced to equivalent cyclic brake applications so that the designer can use the solutions obtained, to ensure that vehicle brakes in general will not overheat and cause a serious loss of brake performance.

This paper describes an investigation of the temperatures developed in automobile braking. The problems considered are discussed under the following two main headings: Part I, Temperature reached in a single brake application to bring the vehicle to rest; and, Part II, Temperatures reached in general motoring.

PART I—TEMPERATURES REACHED IN A SINGLE APPLICATION OF THE BRAKES, TO BRING THE VEHICLE TO REST

Heat generated during braking

In a single brake application, if a vehicle of weight P lb decelerates to rest at a uniform rate a ft/sec², on a level road, under the action of the brakes, then t sec after commencement of braking from an initial speed u ft/sec, the instantaneous speed is $v=(u-at)$ ft/sec. The total braking force acting on the vehicle is Pa/g lb wt, and the total instantaneous energy dissipated at the interface of drum and lining is Pav/g ft-lb/sec, where g is the acceleration due to gravity and is 32·2 ft/sec². The heat generated at this rate of working is Pav/gJ B.Th.U/sec, where J is the mechanical equivalent of heat and is 778 ft-lb/B.Th.U. Depending on the design of the vehicle, some proportion y of the total braking force will be transmitted to the front axle — y is generally 0·6 for an automobile and 0·4 for a lorry or public service vehicle. Thus, if σ is the proportion of the heat generated that enters a drum, the heat inflow in B.Th.U/ft²-sec is

$$\frac{\sigma yPav}{2gA_1J}=\frac{\sigma yPau}{2gA_1J}\left(1-\frac{a}{u}t\right)=\frac{E_1}{T}\left(1-\frac{t}{T}\right)=N_1\,(1-Mt)\quad\ldots\ldots(1)$$

into unit area of a front drum, and

$$\frac{\sigma(1-y)Pav}{2gA_1J}=\frac{\sigma(1-y)Pau}{2gA_1J}\left(1-\frac{a}{u}t\right)=\frac{E'_1}{T}\left(1-\frac{t}{T}\right)=N'_1\,(1-Mt)\quad\ldots\ldots(2)$$

to unit area of a rear drum. The inflow to the linings is

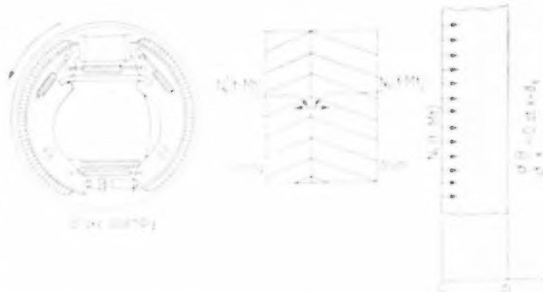


Fig. 1. Development of the problem of heat conduction during a single application of the brakes to bring the vehicle to rest. At the interface, a thermal flux $N_1(1-Mt)$ enters the drum and $N'_1(1-Mt)$ enters the lining, as indicated in the centre diagram—in the mathematical analysis, the curvature of the drum and lining is ignored. With regard to the diagram on the right, it is assumed that there is no loss to atmosphere at d_1, and the heat flow problem finally reduces to one of conduction through infinite slabs, of the drum and lining, between parallel boundaries

$$\frac{(1-\sigma)yPua}{2gA_2J}\left(1-\frac{a}{u}t\right)=\frac{E_2}{T}\left(1-\frac{t}{T}\right)=N_2\,(1-Mt)\quad\ldots\ldots\ldots(3)$$

for the front linings, and

$$\frac{(1-y)\,(1-\sigma)Pua}{2gA_2J}\left(1-\frac{a}{u}t\right)=\frac{E'_2}{T}\left(1-\frac{t}{T}\right)=N'_2\,(1-Mt)\quad\ldots(4)$$

for the rear linings.

Here, A_1 represents the area swept out by the lining in the two drums, which are assumed to be of equal dimensions, and A_2 is the area of friction material in each drum. The quantity σ has been discussed by Newcomb,[1] who derives the relationship

$$\sigma=\frac{1}{1+\dfrac{A_2}{A_1}\left(\dfrac{K_2\rho_2C_2}{K_1\rho_1C_1}\right)^{\frac{1}{2}}}\quad\ldots\ldots\ldots\ldots\ldots\ldots(5)$$

where K, ρ, C refer to the thermal conductivity, density and specific heat respectively; suffixes 1 and 2 refer to the drum and lining material respectively. With conventional asbestos-resin friction materials sliding against cast iron, $\sigma=1/(1+0{\cdot}09\,A_2/A_1)$ and $\sigma=0{\cdot}95$ when A_2 substends a total arc of lining of 220 deg, which is typical of brake linings. Hence, 95 per cent of the heat evolved during braking enters the drum. In a disc brake where A_2/A_1 is generally about 1/9, the value of $\sigma=0{\cdot}99$ so that the amount of heat transferred to a disc is approximately 1·04 times that entering a drum when braking from the same initial energy.

The terms N_1 and M represent E_1/T and a/u respectively, where $E_1=\sigma yPu^2/2gA_1$ is the thermal equivalent, per unit drum area, of the energy imparted to the front drum, and in braking to rest $u=aT$ where T is the duration of the brake application. The other values of N for the rear drums and linings have a similar interpretation. Thus, if T remains constant whilst the initial energy of the vehicle changes, the values N vary as the square of the initial speed.

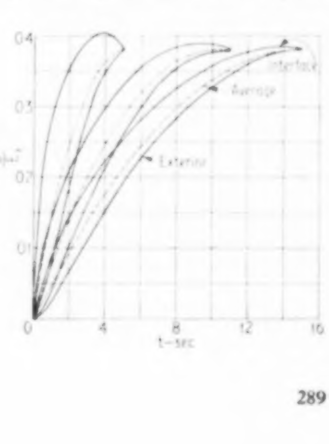


Fig. 2. Variation of the surface temperature θ/E_1 at the interior and exterior surfaces of the drum during 5, 10 and 15 sec applications of the brake, each from identical initial speeds, where $d_1=\frac{1}{4}$ in

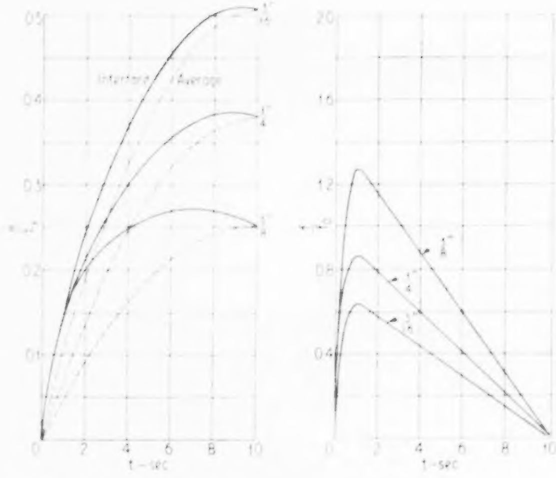


Fig. 3, left. Interface and average temperatures of the drum plotted against time throughout a 10 sec brake application, with drums of thickness $\frac{3}{16}$ in, $\frac{1}{4}$ in and $\frac{3}{8}$ in; in each instance the initial speed of the vehicle before application of the brakes was the same. Fig. 3a, right. Variation with the time of the compressive stress f/f_0 at the friction faces of different drums during a 10 sec application of the brakes

The expression given by equations 1-4 ignores the energy of the rotating parts of the vehicle, the work done due to wheel slip at the road-tyre surface, and the work done against windage and rolling resistance. The rotational energy could be included by addition of an appropriate mass to the weight of the vehicle — rotational energy is generally of the order of 5 per cent of the translational energy. Wheel slip depends on the decelerating force and the tyre coefficient, the latter being slightly dependent on speed. Grime and Giles[2] show that the percentage slip should not exceed 5 per cent for decelerations up to 0·4 g and can be ignored without serious error. An estimation of the deceleration due to windage and rolling resistance can be made by allowing the vehicle to come to a standstill from various speeds when out of gear on a level road and measuring the retardation with a decelerometer. Typical figures are 0·03 g and 0·05 g for a 10 h.p. car running from initial speeds of 30 m.p.h. and 60 m.p.h. respectively. To give the decelerations produced by the brake, this figure can be subtracted from measured decelerations.

Braking down an incline

If a vehicle is descending a long hill, of inclination ϕ, under the action of gravity alone, the accelerating force down the incline is $P \sin \phi$ and, for the vehicle to maintain a steady velocity v ft/sec, a constant braking force of $P \sin \phi$ lb must be applied to the wheels. Hence the instantaneous energy dissipated is $P \sin \phi\ v$ ft-lb/sec, and in this case the heat inflow is

$$N_1{}^* = \frac{\sigma y P v \sin \phi}{2A_1 J} \qquad \ldots\ldots(6)$$

into unit area of a front drum and

$$N'_1{}^* = \frac{\sigma(1-y) P v \sin \phi}{2A_1 J} \qquad \ldots\ldots(7)$$

into unit area of a rear drum.

Similar relationships hold for the heat input to the linings.

Temperatures obtained in a single brake application

Vehicle brake drums in general have radii of curvature which are large compared with their thickness and, consequently, only a small error is introduced in considering a drum to be developed into a flat rectangular plate and in neglecting edge effects, Fig. 1. At any stage during braking, the rate at which heat enters the drum is $N_1(1-Mt)$ at unit area of the friction surface, whilst at the exterior surface of the drum, cooling losses to the atmosphere occur. However, Newcomb[1] has shown that the heat loss by cooling is at the most about 3 per cent of the heat generated during a single brake application and, therefore, can be ignored. Thus, the conduction of heat through a brake drum can be assumed to be one of linear flow through an infinite slab bounded by parallel planes. This heat problem has been solved by Odier and Leutard[3], Bannister[4], Hasselgruber[5], and Newcomb[6-8] and the last author derives the following equations for determining the temperature rise θ, that is, above ambient, at the interface of front drum and lining at any instant t during a brake application of duration T:

$$\theta = \frac{2\alpha_1{}^{1/2} t^{1/2} E_1}{T K_1 \pi^{1/2}} \left(1 - \frac{2}{3}\frac{t}{T}\right) \text{ in the range } 0 < t < 2 \qquad \ldots\ldots(8)$$

and

$$\theta = \frac{\alpha_1 E_1}{T K_1 d_1} \left\{ t\left(1 - \frac{t}{2T}\right) + \frac{d_1{}^2}{3\alpha_1}\left(1 - \frac{t}{T}\right) + \frac{d_1{}^4}{45\alpha_1{}^2 T} \right\} \text{ in the range } 2 < t < T \qquad \ldots\ldots(9)$$

where $\alpha_1 = K_1/\rho_1 C_1$ is the thermal diffusivity of the drum material of thickness d_1.

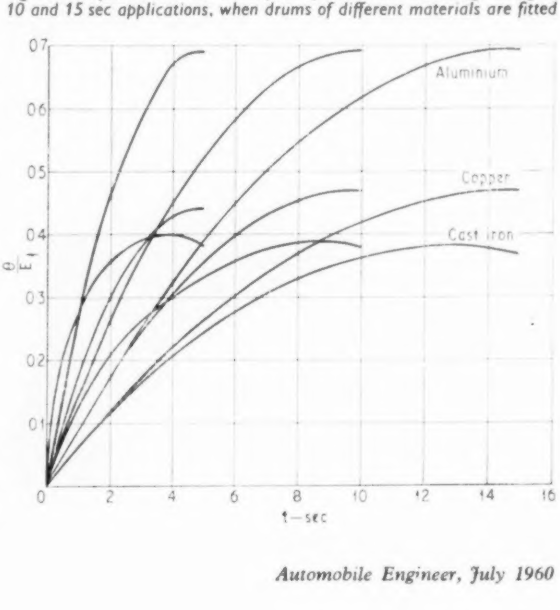
Strictly, these formulae only apply to drums when d_1 lies between $\frac{3}{16}$ and $\frac{5}{16}$ in, and the changeover value $t = 2$ is only approximate, but sufficiently accurate for practical purposes. The thicker the drum and the shorter the brake application the greater the range of time in a brake application in which equation (8) may be used to determine the temperature rise. An exact solution to the heat problem is given in the Appendix.

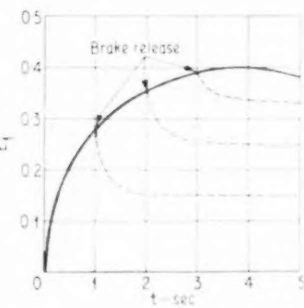
These equations have been verified experimentally by Newcomb[8] to within ± 10 per cent. Furthermore, during the course of a single brake application, the temperature (θ_e) of the exterior surface of the cylindrical part of the drum at any instant is given by

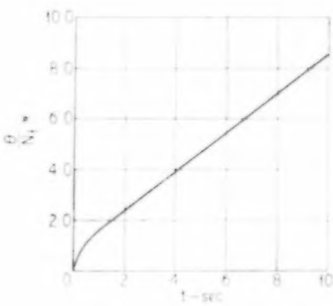
$$\theta_e = \frac{\alpha_1 E_1}{T K_1 d_1} \left\{ t\left(1 - \frac{t}{2T}\right) - \frac{d_1{}^2}{6\alpha_1}\left(1 - \frac{t}{T}\right) - \frac{7 d_1{}^4}{360\alpha_1{}^2 T} \right\} \qquad \ldots\ldots(10)$$

To illustrate the transient nature of the temperature variations, Fig. 2 shows a plot of θ/E_1 against t calculated from equations (8), (9) and (10), using the following data for a typical cast iron drum: $d_1 = 0{\cdot}25$ in $= 0{\cdot}0208$ ft, $K_1 = 0{\cdot}00739$ B.Th.U/ft-sec-deg F, $\rho_1 = 450$ lb/ft^3, $C_1 = 0{\cdot}14$ B.Th.U/lb-deg F, $\alpha_1 = 0{\cdot}000117$ ft^2/sec and $T = 5$, 10 and 15 sec. These curves indicate that in the early stages of braking, an

Fig. 4. Temperatures at the interface of the drum and lining during 5, 10 and 15 sec applications, when drums of different materials are fitted







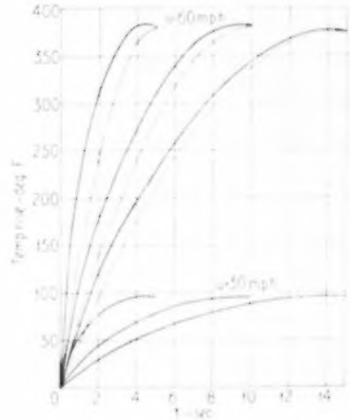


Fig. 5, left. Temperature rise θ/E_1 at the interface when the brake is released after 1, 2 and 3 sec in a brake application of 5 sec. Fig. 6, centre. Variation of the interface temperature θ/N_1 with t when a vehicle is braked at a constant speed down a uniform gradient. Fig. 7, right. Effect of initial speed on the temperature rise of the friction surface of the front brake of a small saloon car, brought to a standstill in different times from the point of application*

extremely rapid increase in temperature occurs at the friction surface and, for the same E_1, the initial gradient is greatest for the shortest stopping time. Also, the temperature attains a maximum near the end of the brake application and this temperature θ_{max} is given by the equation

$$\theta_{max} = \frac{\alpha_1 E_1}{K_1 d_1}\left(\frac{1}{2} + \frac{7d_1^{\,3}}{90\alpha_1^{\,2}T^2}\right) \quad \ldots\ldots (11)$$

The subsequent fall in temperature after θ_{max} is reached results from the vehicle speed having decreased sufficiently to cause the heat inflow to the drum to become less than the rate of conduction away from the friction surface. At the exterior surface of the drum, the temperature rises slowly at first but, as braking continues, it rises at an increasing rate until, at the end of a brake application, little difference in temperature exists between the two surfaces of the drum.

For comparison, the rate of increase in average temperature θ_{av} is also shown in Fig. 2, where θ_{av} is determined from the expression

$$\theta_{av} = \frac{\alpha_1 \int_0^t N(1 - Mt)\,dt}{K_1 d_1} = \frac{E_1 t}{T\rho_1 C_1 d_1}\left(1 - \frac{t}{2T}\right) \quad \ldots\ldots (12)$$

in which it is assumed that all the heat generated up to time t is uniformly distributed throughout the drum. This curve shows that the drum will attain uniform temperature very shortly after the stop, and its value $\Delta\theta$ — when $t = T$ equation (12) — is given by

$$\Delta\theta = \frac{E_1}{2C_1\rho_1 d_1} \quad \ldots\ldots (13)$$

that is

$$\Delta\theta = \frac{\text{thermal equivalent of total K.E. imparted to drum}}{\text{thermal capacity of the drum}}$$

Rear drum temperatures are calculated in a similar manner except that E_1 is replaced by E'_1 in equations (8) and (9).

The effect of varying the drum thickness, when E_1 is the same for all these drums is illustrated in Fig. 3. These curves show the rubbing path and average drum temperatures throughout a 10 sec brake application, for drums of thicknesses of $\frac{5}{16}$ in, $\frac{1}{4}$ in and $\frac{3}{8}$ in, which are commonly used on modern automobiles. It can be seen that a thin brake drum will attain a high temperature on account of its small heat capacity and that the average temperature after braking is inversely proportional to the drum thickness. Furthermore, the excess of surface temperature, relative to average temperature, decreases as the thickness decreases and consequently, the thermal stresses will be reduced by using thinner brake drums.

Newcomb[9] shows that the compressive stress f at the rubbing path in the axial and tangential direction of the drum is the same and is given by

$$f = \frac{\alpha_1^{1/2} f_0}{K_1}\left[\frac{2t^{1/2}}{\pi^{1/2}}\left(1 - \frac{2}{3}\frac{t}{T}\right) - \frac{\alpha_1^{1/2} t}{d_1}\left(1 - \frac{t}{2T}\right)\right]$$

when t is approximately less than 1·5 sec

$$\text{and } f = \frac{\alpha_1 f_0}{d_1 K_1}\left[\frac{d_1^{\,2}}{3\alpha_1}\left(1 - \frac{t}{T}\right) + \frac{1}{45}\frac{d_1^{\,4}}{\alpha_1^{\,2}T}\right] \text{ when } t > 1{\cdot}5 \text{ sec}$$

$$\text{where } f_0 = \frac{E_1 Y l}{T(1 - \nu)}$$

Y = modulus of elasticity.
l = coefficient of linear expansion.
ν = Poisson's ratio.

Fig. 3a shows the variation with time of compressive stress f/f_0 at the rubbing path of a drum of thickness $\frac{5}{16}$ in, $\frac{1}{4}$ in and $\frac{3}{8}$ in, when the duration of the brake application is 10 sec. These curves show that the maximum stress is proportional to the drum thickness. In a practical case, consider a car weighing 3,000 lb, having a drum swept area of 84·8 in^2 with a drum thickness $\frac{1}{4}$ in and made of cast iron, where $Y = 1{\cdot}8 \times 10^7$ lb/in^2, $\nu = 0{\cdot}26$ and $l = 0{\cdot}0000055$ per deg F. If this car is braked from 120 m.p.h. to rest in 10 sec, corresponding to a rate of deceleration of 0·54 g, the maximum compressive stress as determined from the above equations is 21,500 lb/in^2. This figure is well below the compressive strength of cast iron and, therefore, under heavy duty applications the bulk stresses are unlikely to cause cracking or failure of the drum when uniform contact occurs between the drum and lining.

The effect of changing the thermal properties of the drum is shown in Fig. 4. Here θ/E_1 is plotted against t for 5, 10 and 15 sec applications, when drums of different materials, of thickness $\frac{1}{4}$ in, are braked to rest; for the drums of aluminium and of copper, $K_1 = 0{\cdot}0323$ B.Th.U/ft-sec-deg F, $C_1 = 0{\cdot}206$ B.Th.U/lb-deg F, $\alpha_1 = 0{\cdot}000931$ ft^2/sec and $K_1 = 0{\cdot}0625$ B.Th.U/ft-sec-deg F, $C_1 = 0{\cdot}0914$ B.Th.U/lb-deg F, $\alpha = 0{\cdot}001225$ ft^2/sec respectively. For comparison, temperatures for a cast iron drum are also shown. These curves indicate that copper, which has a high thermal conductivity, is useful in reducing the rate at which the interface temperature rises, particularly for brake applications of short duration. However, its thermal capacity is slightly lower than that of iron so that, ultimately, a thin copper drum will attain a higher temperature than an iron drum. Likewise, with aluminium, which has a higher thermal conductivity than iron, there is a slightly lower rate of temperature rise after the start of braking, but eventually, because of its low thermal capacity, high thermal conductivity is a less significant factor and aluminium attains a higher temperature than iron.

The foregoing equations giving the temperature rise are based on braking the vehicle to a standstill. Generally the

vehicle is braked from an initial velocity u to a final velocity U, with a duration of braking t_1 sec, where $0 < t_1 < T$. For the case where the brake is released before the vehicle comes to rest, the method of calculating the temperature has been considered by Bannister[5]. The results are best discussed for a typical application, for example, in respect of a brake that is released after 1, 2 and 3 sec, when the total stopping time of a vehicle would be 5 sec, Fig. 5. The curves indicate that up to t_1 the interface temperature follows exactly the shape it would do when braking to rest. After t_1, however, there is a rapid decrease in surface temperature, which levels off near $t = T$, that is, approaching what would be the total stopping time of the vehicle if it were allowed to come to a standstill at the same rate of deceleration. In these circumstances the final average temperature rise of a front drum in the absence of cooling is

$$\Delta\theta = \frac{\sigma y P(u^2 - U^2)}{4gA_1 J \rho_1 C_1} \quad \ldots\ldots\ldots\ldots (14)$$

that is

$$\Delta\theta = \frac{\text{thermal equivalent of total K.E. imparted to drum}}{\text{thermal capacity of the drum}}$$

During an application, the friction surface of a brake lining is at the same temperature as that of the drum, as given by equations (8) and (9), but since the lining is a poor thermal conductor, there is a steep temperature gradient across it when braking commences. Consequently, heat will, therefore, be conducted slowly through the lining and then to the metal shoe, whose thermal capacity cannot be ignored, until the lining and shoe are at a uniform temperature $\Delta\theta'$. This is reached some 5 to 10 sec after the end of the application, and $\Delta\theta'$ is given by

$$\Delta\theta' = \frac{(1-\sigma) y P u^2}{8gJW_3C_3} \quad \ldots\ldots\ldots\ldots (15)$$

provided the heat loss in this time is negligible. The quantity W_3C_3 is the combined thermal capacity of lining and shoe. Since much more heat goes into the drum than into the lining, because the thermal conductivity of the latter is so much smaller, the average temperature of the lining and shoe is less than that of the drum. The ratio $\Delta\theta'$ to $\Delta\theta$ is

$$\frac{\Delta\theta'}{\Delta\theta} = \frac{(1-\sigma)\, A_1 C_1 \rho_1 d_1}{2W_3 C_3 \sigma} \quad \ldots\ldots\ldots\ldots (16)$$

which has the value of approximately 0·2 for a typical brake assembly. Thus, in general, the average lining and shoe temperature is approximately one fifth that of the drum after one brake application.

If the vehicle is travelling down a gradient and the brake is steadily applied to maintain a constant speed, then $M = 0$

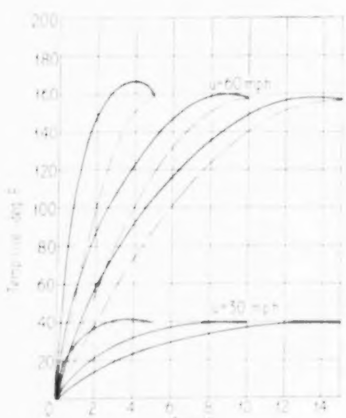


Fig. 8. Effect of initial speed on the temperature rise of the friction face of the brake drum of a high power car brought to a standstill in three different times

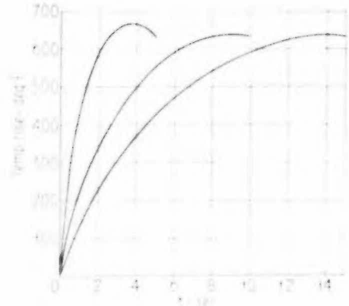


Fig. 9. The temperature rise at the friction surface of the front brake drum of a high power car brought to a standstill in different times from a high initial speed of 120 m.p.h.

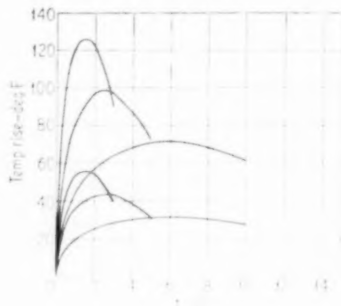


Fig. 10. Friction surface temperature rise in the rear brake drum of a public service vehicle brought to a standstill with brake applications of 3, 5 and 10 sec from initial speeds of 20 m.p.h. and of 30 m.p.h.

and from the previous equations the temperature of the rubbing path is

$$\theta = \frac{2\alpha_1^{1/2} t^{1/2} N_1^*}{K_1 \pi^{1/2}} \text{ in the range } 0 < t < 2 \quad \ldots\ldots\ldots (17)$$

and

$$\theta = \frac{\alpha_1 N_1^*}{K_1 d_1}\left\{t + \frac{d_1^2}{3\alpha_1}\right\} \text{ when } t > 2 \quad \ldots\ldots\ldots (18)$$

Fig. 6 shows the variation of θ/N_1^* with time for ¼ in thick cast iron drum; initially the temperature varies as the square root of the time, and then it increases at a linear rate. This curve shows that in long brake applications down a hill the temperature may reach very high values and affect the brake performance.

Temperatures attained in various types of vehicles and of brake applications

The curves mentioned so far illustrate the general characteristics of the formulae for obtaining the temperature rise at the rubbing path, but now, actual values of temperature will be determined for different types of vehicles braked from different initial energies. It is assumed that all the energy is dissipated in the form of heat at the brakes. The cases considered are the front drum temperatures in a small saloon car and in a high-power car, and the rear drum temperatures in a public service vehicle.

In the front brake of a typical small saloon car of weight 1,624 lb, $y = 0{\cdot}6$, $\sigma = 0{\cdot}95$, $d_1 = \frac{3}{16}$ in $= 0{\cdot}0156$ ft and $A_1 = 0{\cdot}191$ ft². When this car is braked from speeds of 30 m.p.h. and 60 m.p.h. to rest, in times of 5, 10 and 15 sec, the temperatures at any instant during each stop, as calculated from equations (8) (9), are shown in Fig. 7. The temperatures attained for a high-power car of weight 2,772 lb, $y = 0{\cdot}6$, $\sigma = 0{\cdot}95$, $d_1 = 0{\cdot}0208$ ft and $A_1 = 0{\cdot}59$ ft², when braked from 30, 60 and 120 m.p.h. in times of 5, 10 and 15 seconds, are shown in Figs. 8 and 9. For the case of the fully laden public service vehicle of weight 12 tons, where $1 - y = 0{\cdot}6$, $d_1 = 0{\cdot}0416$ ft and $A_1 = 2{\cdot}095$ ft², when the vehicle is braked from 20 to 30 m.p.h. to rest in times of 5, 10 and 15 sec, the temperatures are shown in Fig. 10. The rates of decelerations corresponding to these speeds and times of braking are shown in Table I. Since the deceleration for most normal brake applications is approxi-

TABLE 1—Rates of deceleration at different speeds

Speed, m.p.h.	*Deceleration, g* 5 *sec*	10 *sec*	15 *sec*
20	0·18	0·09	0·06
30	0·27	0·135	0·09
60	0·54	0·27	0·18
120	1·08	0·54	0·36

mately 0·1 g, most of the values in the Table are equivalent to emergency stops.

These curves show that within two or three seconds, friction surface temperatures of over 400 deg F may be developed during braking at high speeds. Brakes of small cars can still attain high temperatures because of their thin brake drums, but in this case the low value of the maximum car speed will limit the temperature, which is not likely to be much in excess of 400 deg F in one application provided the braking path area, of present day vehicles, is not changed. However, with heavy, high power cars, temperatures can be produced that may cause physical changes in the brake components. It is also of interest that the effect of varying the rate of deceleration is to change the initial rate of increase in temperature but to cause little change in the maximum temperature. This is because the cooling losses in such a short time can be neglected.

In the case of the public service vehicle, on account of its thick brake drums, equation (8) applies to the temperature rise for the whole of the duration of braking provided the stopping time is five seconds or less. In these short time applications the temperature increases rapidly and attains a maximum value halfway through the brake application and, as shown by Odier[10], the value $\theta_{max} = \sqrt{2}\theta_T$, where θ_T is the interface temperature rise at the end of the stop. For long applications of the brakes, the behaviour in respect of θ is similar to that for thin brake drums.

If the high-power car is being steadily braked so as to maintain a constant speed of 30 m.p.h. down a 1 in 12 incline for a distance of 0·75 mile, then the temperature variation with time of braking is as shown in Fig. 11. This curve shows that high temperatures in the region of 500 deg F may be developed during continuous braking down long steep inclines.

Temperatures reached in different brakes—drum or disc

The equations given previously apply to a drum type brake, but in view of the increasing use of disc brakes it is important to know whether a disc brake attains the same temperatures as a drum type when braking under identical conditions. As is well known, a disc brake consists of two blocks of friction material, or pads, which are pressed one against each side of a rotating annulus, usually made of ferrous material, the latter being rigidly attached to the vehicle whose motion is to be retarded. Thus, since there are two friction surfaces to a disc, if one applies the principle of symmetry and divides the disc into two, the heat problem becomes identical to that of a drum brake. Hence, if half the disc thickness is denoted by d, then equations (8) and (9) will still determine the friction surface temperature except that E_1 is replaced by E_1^* owing to the change in value of σ. As shown previously the value of σ for a typical disc brake is 1·04 that of a drum brake. Thus, for the same rubbing path area, the interface temperature rise in a single brake application is approximately 4 per cent higher than that of a drum brake, provided braking takes place under similar conditions. Experimental measurements of disc temperatures with a radiation pyrometer are shown in a paper by Newcomb[11] and these results agree, to within 15 per cent, with the theoretically calculated values.

The foregoing analysis ignores cooling losses but since, in a disc brake, a greater area of disc surface is exposed to the atmosphere than in a drum brake, it might be thought that this will have a considerable effect on the temperature rise. Below a temperature of 600 deg C, cooling is mainly by convection and these losses have been considered by Newcomb[12], who concludes that the surface temperature in a single brake application will be reduced by approximately 6 per cent in an air stream of high velocity. However, in modern practice most of the disc surface is shrouded by the wheel or by covers to prevent dust from the road surface contaminating the disc, so little use is made of this additional cooling area. The overall effect is that both drum and disc brakes operate at much the same temperatures under comparable conditions.

By the use of cerametallic pads of higher heat conductivity than conventional materials, the heat transferred to the disc can be reduced to $\sigma = 0·95$ for typical automotive pad sizes, and so will only reduce the surface temperature by approximately 5 per cent; the effect of pad material is discussed by Newcomb[12].

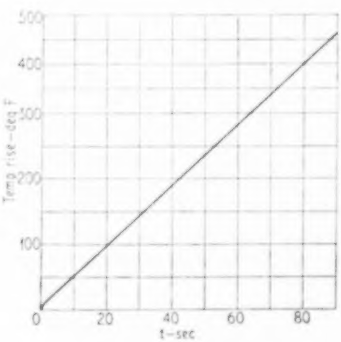
The main advantage of disc brakes over drum brakes is that, at high temperatures, the disc expands towards the pad, thus causing no loss in pedal travel and enabling the disc to operate at a higher temperature before thermal stresses and distortion become too pronounced. Another advantage is that, if the volume wear rate in a pad is the same as in a drum lining, the pad, since it is smaller in area, will exhibit a greater loss of thickness and this tends to prevent a build up of surface films on it.

Comparison of cast iron with bimetallic brake drums

In most automobiles the brake drums are of cast iron having a total carbon content of 3·00-3·50 per cent and 0·5-0·8 per cent combined carbon, with some additions of chrome nickel and a fairly small percentage of Si and Mn, yielding a Brinell Hardness of 200-240. However, on racing and many Continental and American cars, the brake drums are in many instances composed of two parts, an inner cylinder which is thin and made of cast iron and a much thicker outer layer of light material with a high thermal conductivity, for example, aluminium alloy. The outer portion is generally finned. A cast iron layer is essential because of its resistance to scoring, maintenance of strength at elevated temperatures, stable coefficient of friction, ability to resist deformation, resistance to heat checking and long life in normal usage. The presence of the thick layer of high conductivity material causes heat to be conducted away more rapidly than with cast iron alone and, accordingly, the maximum temperature attained should be lower than with ordinary brake drums of the same weight.

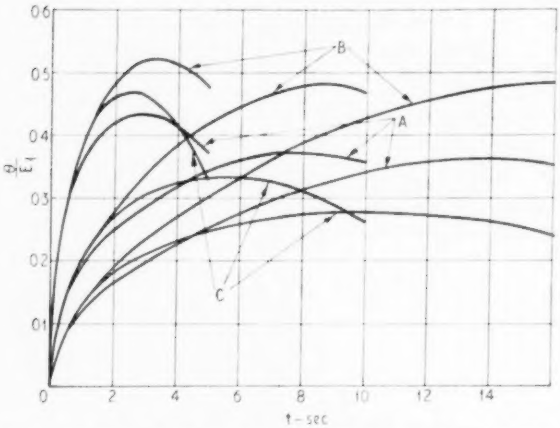
The problem of the flow of heat in a bimetallic drum is similar to that in a single medium, in that at one surface there exists a thermal flux $N(1 - Mt)$ entering the cast iron

Fig. 11. Temperature rise at the friction surface in the front brake drum of a high power car being braked steadily down a 1 in 12 gradient, while the vehicle speed is maintained at 30 m.p.h. for a distance of 0·75 miles



layer and that there is no flow of heat at the outer surface. However, the presence of the interface of cast iron and aluminium affects the solution to the problem, because of the different thermal properties of the materials. This heat problem has been considered by Newcomb[15], who derives in equations (13) and (14) of that paper a solution to permit calculation of the transient temperature at any depth in both media of the drum.

A typical bimetallic drum has a cast iron layer of 0·1181 in



A bi-metallic drum; **B** cast iron drum of equivalent weight; **C** cast iron drum of equivalent volume

Fig. 12. Variations of θ/E_1 at the interface of the drum and lining during 5, 10 and 16 sec brake applications with different drums, as in the key

(0·3 cm) and an aluminium layer of 0·4724 in (1·2 cm), with a thermal conductivity of 0·0323 B.Th.U/ft-sec-deg F, a density of 171 lb/ft^3 and a specific heat of 0·206 B.Th.U/lb-deg F. For a front bimetallic drum, values of the temperature rise θ/E_1 at the rubbing path have been determined from the above equations at various times during a 5, 10 and 16 sec brake application to bring the vehicle to rest. These, together with values of θ/E_1 at the interface of drum and lining, for a cast iron drum of equivalent weight and of equivalent volume to that of the bimetallic drum, are shown graphically in Fig. 12.

The curves in Fig. 12 show that the maximum temperature rise of a bimetallic drum is lower than that of a drum of equivalent weight made of cast iron. In particular, the advantage of using a bimetallic drum becomes more pronounced as the time of brake application increases, that is, when braking from higher initial speeds. For example, the ratios of the maximum temperatures reached by the cast iron drum to those of the bimetallic drum of equivalent weight are 1·13 : 1, 1·29 : 1 and 1·33 : 1 at 5, 10 and 16 sec stopping times respectively. If a cast iron drum of equivalent volume to that of the bimetallic drum is used, a marked difference results; the cast iron drum reaches a lower maximum temperature, and this becomes more noticeable with increase in duration of the brake application. This is owing to the relative physical properties of the two metals: although the aluminium has a high thermal conductivity compared with cast iron, its thermal capacity—product of density and specific heat—is much lower and, consequently, the cast iron drum attains a lower average temperature. However, though the cast iron gains on an equal volume basis it must be emphasized that its weight will be approximately twice that of a bimetallic drum, and since this is unsprung weight, it is an important factor in respect of modern cars, especially racing cars.

Thus, there is some advantage in using bimetallic drums, as opposed to cast iron drums, provided they are used on an equal weight basis. However, owing to the thermal expansion of aluminium being twice as great as that of cast iron, failure of the aluminium-iron bond may occur at high temperatures. These temperatures can be reduced to some extent by casting cooling fins in the aluminium body or alternatively casting the drum as an integral part of the wheel and ensuring adequate forced convection around the drum. At various depths inside the bimetallic drum steep temperature gradients still exist in the cast iron layer so that high thermal stresses and surface cracking can still be encountered under adverse conditions.

CONCLUSIONS

Part I

i. Of the amount of heat generated during braking, with conventional materials, approximately 95 per cent goes to the drum, in drum brakes, and 99 per cent to the disc when disc brakes are employed.

ii. Extremely rapid increases in temperature may be produced during braking. With thin drums, the metal surrounding the rubbing path is almost at a uniform temperature at the end of the brake application and this value is not much lower than the maximum temperature reached during the application.

iii. The effect of reducing the drum thickness during braking, with the same rubbing path size and initial vehicle energy, is to increase the maximum temperature reached but reduce the thermal stresses. The average temperature attained is inversely proportional to the drum thickness.

iv. After one brake application, the uniform temperature of a brake lining and shoe is generally less than a fifth of that attained by the drum.

v. A drum material having a high thermal conductivity will reduce the rate of increase in temperature at the friction surface in a single brake application, particularly when the braking time is of short duration. However, unless the thermal capacity is also high, the average temperature attained will be high and, for this reason, cast iron is a better drum material than the high conductivity materials copper and aluminium.

vi. Disc brakes operate at much the same temperatures as drum brakes provided the conditions are comparable, that is, the same area of rubbing path, the mean diameter of disc is equal to the internal drum diameter, the disc thickness is twice the drum thickness and each is braked from identical initial energies.

vii. Bimetallic drums are useful in reducing the maximum temperature attained, as compared with cast iron drums of the same weight. For example, the ratio of the maximum temperature reached by the cast iron drum to that of the bimetallic drum of equivalent weight increases from 1·13 : 1 to 1·33 : 1 as the stopping time changes from 5 to 16 sec. However, it is inadvisable to use bimetallic drums above a certain temperature, because of the differential rates of expansion of aluminium and cast iron.

APPENDIX TO PART I

Transient temperature rise at the friction surface

The exact solution giving the temperature rise in a front drum at any time t throughout a brake application can be expressed in the following ways, the first, derived by Newcomb[6], stating that

$$\theta = \frac{E_1 \kappa_1^{1/2}}{K_1 T}\left\{\frac{2t^{1/2}}{\pi^{1/2}}\left(1-\frac{2}{3}\frac{t}{T}\right) + 4t^{1/2}\sum_{n=1}^{\infty} ierfc\,\frac{nd_1}{(\kappa_1 t)^{1/2}} - \frac{16t^{3/2}}{T}\sum_{n=1}^{\infty} i^3erfc\,\frac{nd_1}{(\kappa_1 t)^{1/2}}\right\} \qquad (40)$$

$$\text{where} \quad i^n erfcz = \int_z^\infty i^{n-1}\, erfcy\, dy, \quad n = 1, 2, 3 \ldots$$

$$\text{with} \quad i^0 erfcz = erfcz = \frac{2}{\pi^{1/2}} \int_z^\infty \exp(-y^2)\, dy$$

and the above functions are given in tabular form by Carslaw and Jaeger[14].

The other solution, due to Odier and Leutard[3] and Hasselgruber[5], is

$$\theta = \frac{\alpha_1 E_1}{d_1 K_1 T}\left[t\left(1 - \frac{t}{2T}\right) + \frac{d_1^2}{3\alpha_1}\left(1 - \frac{t}{T}\right) + \frac{1}{45}\frac{d_1^4}{\alpha_1^2 T} - \frac{2d_1^2}{\alpha_1 \pi^2} \sum_{n=1}^{\infty} \frac{1}{n^2} \exp\left(-\frac{\alpha_1 n^2 \pi^2 t}{d_1^2}\right) - \frac{2d_1^4}{\alpha_1^2 \pi^4 T} \sum_{n=1}^{\infty} \frac{1}{n^4} \exp\left(-\frac{\alpha_1 n^2 \pi^2 t}{d_1^2}\right)\right] \quad (41)$$

Equation (40) is particularly suitable for thicker drums, since the series rapidly converges for higher values of $d_1/(\alpha_1 t)^{1/2}$. The solution (41) is more convenient for thin drums for, provided $d_1/(\alpha_1 t)^{1/2}$ is approximately less than 1, that is, t approximately greater than 2, the exponential series can be ignored.

If the exponential series in equation (41) can be neglected, the maximum value of θ is

$$\theta_{max} = \frac{\alpha_1 E_1}{K_1 d_1}\left(\frac{1}{2} + \frac{7d_1^4}{90\alpha_1^2 T^2}\right) \quad (42)$$

and if the error integral terms of equation (40) can be ignored, the maximum value of θ is

$$\theta'_{max} = \frac{0{\cdot}53\, E_1 \alpha_1^{1/2}}{K_1 T^{1/2}} \quad (43)$$

$$\text{The ratio of } \frac{\theta_{max}}{\theta'_{max}} = Z = \frac{1}{1{\cdot}06\lambda}\left(1 + \frac{7}{45}\lambda^4\right) \quad (44)$$

where $\lambda = d_1/\alpha_1^{1/2} T^{1/2}$. Equation (44) has a minimum when $\lambda^4 = 15/7$, that is, $\lambda = 1{\cdot}21$. Hence, for values of $\lambda < 1{\cdot}21$, θ_{max} exceeds θ'_{max}; when values of λ exceed 1·21, the simplified version of equation (41) becomes $\theta_{max} = \theta'_{max}$.

References

1. T. P. NEWCOMB: "Determination of the Area of Friction Surfaces of Automotive Vehicles", *Proc. I. Mech. E. (Automobile Division)*, to be published
2. G. GRIME and C. G. GILES: "The Skid-resisting Properties of Roads and Tyres", *Proc. I. Mech. E. (Automobile Division)* p. 19, 1954-55
3. J. ODIER and P. LEUTARD: "Etude des phénoménes thermiques de surface dans les problèmes de frottement appliqués au freinage", *Bulletin Société Française des Mécaniciens* (1er trimestre) No. 23, p. 23, 1957
4. F. K. BANNISTER: "Transient Temperatures in Racing-car Brake Drums", *Engineering*, Vol. 183, No. 4748, p. 304, 1957
5. H. HASSELGRUBER: "Die Berechnung der Temperaturen an Reibungskupplungen", *Dissertation Techn. Hochschule, Aachen*, 1953
   See also "Temperaturberechnungen für Mechanische Reibkupplungen", *Friedr. Vieweg and Sohn*, Braunschweig 1959
6. T. P. NEWCOMB: "The Flow of Heat in a Parallel-faced Infinite Solid", *Brit. J. Appl. Phys.*, Vol. 9, p. 370, 1958
7. T. P. NEWCOMB: "Flow of Heat in a Composite Solid", *Brit. J. Appl. Phys.*, Vol. 10, p. 204, 1959
8. T. P. NEWCOMB: "Transient Temperatures in Brake Drums and Linings", *Proc. I. Mech. E. (Automobile Division)*, 1959
9. T. P. NEWCOMB: "Thermal Stresses in Brake Drums", *Engineering*, Vol. 189, No. 4,899, p. 369, 1960
10. J. ODIER: "Contribution à l'étude de l'influence de la température sur les problèmes de freinage", *C.R. Acad. Sci.*, Paris, Vol. 238, p. 1288, 1954
11. T. P. NEWCOMB: "Transient Temperatures attained in Disc Brakes", *Brit. J. Appl. Phys.*, Vol. 10, p. 339, 1959
12. T. P. NEWCOMB: "Temperatures reached in Disc Brakes", *Proc. I. Mech. E. (Automobile Division)*, to be published
13. T. P. NEWCOMB: "Analysis of Brake Drum Temperatures in Racing Cars", *Engineering*, Vol. 187, No. 4864, p. 709, 1959
14. H. S. CARSLAW and J. C. JAEGER: "Conduction of Heat in Solids", 2nd Ed., p. 485, Oxford, *Clarendon Press*, 1959

Part II of this article, Temperatures Reached in General Motoring, will be in the next issue of Automobile Engineer.

Long Ferodo History

FEW manufacturers of components can claim so long an association with the automobile industry as can Ferodo Ltd., the world-famous manufacturers of friction materials. Herbert Frood, the founder of the firm, began his experiments on these materials in 1897, with a view to improving the rudimentary brakes then fitted to horse-drawn vehicles. Four years later, his first patent in this connection was accepted and manufacture of brake blocks began in Manchester. In 1904, after a brief trial, the London General Omnibus Co. adopted Frood's brake linings for its experimental motor buses.

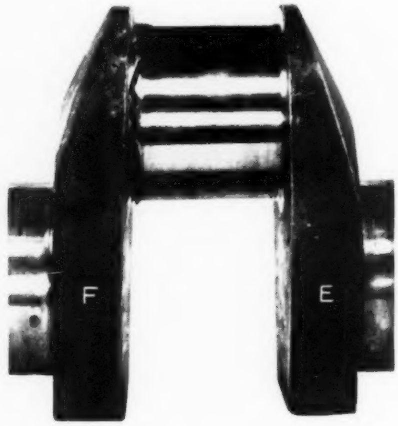
The next important step forward was made in 1908—by which time the company had moved to Chapel-en-le-Frith—with the introduction of woven asbestos as a basis for friction linings. Asbestos-base materials were, of course, able to withstand far higher temperatures than could those previously used, which had a cotton base. Some four years afterwards, die-pressed linings were introduced. Early in World War I, intensive research resulted in the evolution of a new material, known as 9L, which combined good resistance to heat and oil with a physical strength considerably greater than that of its predecessors. Because of the war, and the difficult times that followed, this material could not be made generally available until 1922.

During the next sixteen years, steady progress was made with woven friction linings; several new types were introduced as the demands made on brakes and clutches became more severe. Another milestone was reached in 1938, when moulded materials, of high performance, were first developed. Again, war interfered with normal progress, and it was not until 1947 that this new type was marketed for cars and commercial vehicles.

With the advent of disc brakes, much effort has been devoted since 1953 to the development of satisfactory materials for them—the requirements of these brakes differ markedly, of course, from those of drum brakes. Finality has certainly not yet been reached in this particular field. During the last few years, though, the drum brake has not been neglected, and last year the high-friction AM series of linings was introduced for commercial vehicles. These linings were described in the September 1959 issue of *Automobile Engineer*.

Crankshaft Failure

Report on the Fatigue Fracture of an Engine Crankshaft, the Journals of Which Had Been Surface Treated by Welding Deposition and Chromium Plating



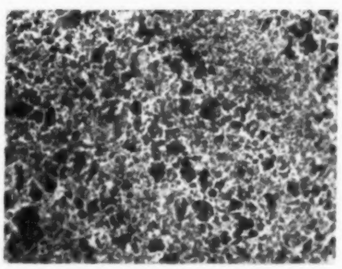
SOME reports on fatigue failures have been collected by the International Institute of Welding, of 32, Bd. de la Chapelle, Paris 18. Among them is one on the fatigue failure of the crankshaft of a large V-12 diesel engine with an output of 500 b.h.p. This crankshaft was a perfectly sound steel forging with a suitable grain structure. However, etching with iodine showed that the crankpin, where failure occurred, had been surface treated by welding deposition and subsequently chromium plated. The plating itself did not extend as far as the fracture area. A chemical analysis of the base metal in the vicinity of the fracture revealed the following content: C 0·36 per cent, Si 0·24 per cent, Mn 1·16 per cent, Ni 0·84 per cent, Cr 0·07 per cent, Mo 0·04 per cent, S 0·032 per cent, P 0·021 per cent, and Cu 0·16 per cent.

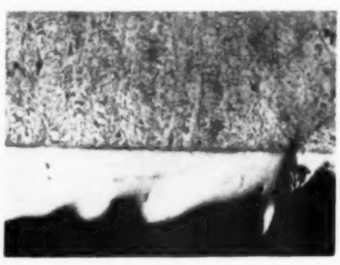
The fracture had originated at several points, all of which were on the fillet connecting the pin to the web at one end of number 3 crankpin. It was in a plane approximately parallel to the adjacent face of the crankweb. At this point the pin was completely severed, but there was also a crack at the other end of the same pin. The diameter of the pin was 6·5 in. There was no perceptible deformation of the pin and the grain at the fracture was characteristic of progressive failure, without, however, being very fine. It is reported that there were no abnormal circumstances leading to the fracture, and, in particular, the alignment of the crankshaft was satisfactory.

The broken crank, by virtue of its position in the central portion of the crankshaft, transmitted only a limited torque. However, the action of the connecting rod caused it to be loaded with a cyclic bending force of normal magnitude. The operational speed of the engine was 1,000 r.p.m., and complete fracture occurred at 114 hours of operation after the crankshaft had been taken from stock and put into service. The amplitude of stress was not measured.

It is not known under what conditions the surface treatment was carried out. An examination showed that the deposition affected the whole of the surface of the pin and

Above. The left-hand of these two illustrations shows the surface of the fracture, while the right-hand one is of the complete crank, with the fracture in a plane adjacent to the web F on the left





The three illustrations on the right depict the differences in structure observed over the cross section of the pin of number 3 crank, as shown at the top of the opposite page. The surface was mechanically polished, picral etched for 30 sec, and reproduced at a magnification × 63. The structure in the upper diagram is that of the unaffected zone of parent metal; it is just possible to detect bands corresponding to the direction of forging grain. In the central diagram, the heat affected zone of the parent metal is at the top and the deposited metal at the bottom, while the lower diagram is of the surface zone

Of the two illustrations on the right, the left-hand one is a section through the crank throw E illustrated on the opposite page, and the other is of part of the throw F—the white areas are the weld metal

extended on to the fillets joining it to the webs. The thickness of metal deposited on the cylindrical surface after it was machined was about 0·177 in, but was less on the fillets.

It is assumed that failure was caused by the combined effects of the stress concentration in the fillet and the weakening due to the surfacing. It is concluded that surface reclamation of shafts by welding deposition can cause a significant reduction in the service life of a shaft subjected to dynamic stressing. Many other failures of this kind have been experienced, and in view of the magnitude of this problem, Commission XIII of the Institute has inaugurated a co-operative investigation, with the object of obtaining a better understanding of the influence of surface deposition and of the various factors involved. Representative organizations in a number of countries are participating in the investigation, and eventually a joint report will be issued.

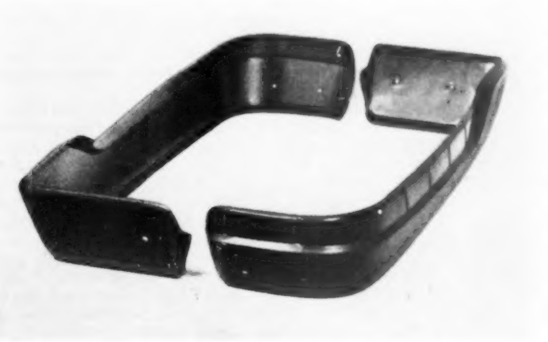
Plastics on American Fords

SAVINGS of 30 per cent in component cost and 2·8 lb in weight have been effected by the change from steel to linear polyethylene for the seat side shields of the American Ford 1960 cars. The linear form of polyethylene, or polythene as it is more commonly called in Britain, has the same chemical composition as the normal form but the molecular arrangement is more orderly. In consequence, most of its physical properties are superior to those of ordinary polythene. It is more dense, and its strength and rigidity are considerably higher; it is also more resistant to solvents, is less permeable and has a higher softening temperature.

The brand used by Ford is Marlex, a product of Phillips Chemical Company, and development of the new side shields took ten months. They are produced by injection moulding and are stated to have the following advantages over the superseded components: lower piece and tooling costs, better styling, reduced weight, and increased resistance to scuffing and denting. According to Ford, the potential styling benefit was the primary reason for the decision to investigate the use of plastics for this application, and linear polythene was selected as the most suitable material.

In the making of the prototype parts, the benefits of convenience and economy became apparent. The samples were, in fact, produced by vacuum forming embossed sheet stock over plaster moulds, in which minor variations could readily be embodied. After the prototype tests proved successful, a production mould was made in seven weeks; a further two weeks covered the making and testing of production prototype shields.

The seat side shields of linear polythene fitted to 1960 Ford cars



For the following few months, a dual production programme was carried out on the 1960 components, to enable a comparison to be made between shields of metal and those of Marlex. This programme confirmed the economies effected by the use of the plastics material. The self-coloured, injection-moulded shields were made in a single operation, whereas those of steel required several manufacturing operations, followed by cleaning, priming and painting.

It is expected that Ford's success with this relatively new material will encourage a rapid increase in its use by other companies. Marlex is already employed for minor components on 1960 Chevrolet, Chrysler and Mercury cars, and on Dodge trucks. It should not be thought, however, that the production and application of linear polythene is confined to the U.S.A. The material is also manufactured in this country and the desirability of using it for this type of application is at present being investigated by a number of British vehicle manufacturers.

Standard for Gear Hobs

IT HAS recently been found necessary to revise the British Standard on gear hobs, first published in 1953. Part 1 of the revised standard, B.S.2062 : 1959, is now available, and is noteworthy in several respects. The specification covers the range of diametral pitches from 1 to 20 only, and the tooth form and pitches detailed are in accordance with the International Standards Organization's recommendations R.53 and R.54. Also, a new high-precision hob, comparable with the grade AA for marine gear hobs, is included in the specification, since hobs of this degree of accuracy have been found necessary for producing the gears needed in a number of applications.

Recommended limiting dimensions for parallel arbors are given in an appendix, to enable the correct fit between hob and arbor to be obtained. Included with the tables of permissible errors are typical methods of testing, the presentation being similar to that adopted for the recently introduced British Standards for gem cutting tools. Copies of B.S.2062 : 1959 are obtainable, price 8s 6d each, from the British Standards Institution, Sales Branch, 2, Park Street, London, W.1. Postage is charged extra to non-subscribers.

Recent Publications

Brief Reviews of Current Technical Books

Basic Electricity

London: THE TECHNICAL PRESS LTD., 1 Justice Walk, Chelsea, S.W.3. 1959. $10 \times 6\frac{1}{4}$. Five volumes, price 12s. 6d. per volume, 55s. per set.

In this work are presented the essentials of a course devised some years ago at the request of the U.S. Navy, its five parts representing a course of technical training in basic electricity. A special, electronics training investigation team of the Royal Electrical and Mechanical Engineers undertook the work of adapting the original manuals to British usage and terminology, and the revised edition has been adopted as a basic work for the training of R.E.M.E. technicians.

Some idea of the scope of the work can be obtained from its contents, which are as follows. Part 1 contains: What electricity is; How it is produced; Primary and secondary cells; Magnetism and magnetic fields; Current flow; Moving coil meter; Voltage; Resistance. In Part 2 the following are dealt with: What a circuit is; D.C. series circuits; Ohm's law; Power in D.C. circuits; D.C. parallel circuits; Kirchhoff's laws. The headings of the sections in Part 3 are: What alternating current is; Resistance, Inductance, Capacitance in A.C. circuits; Power in A.C. circuits. Part 4 has five sections entitled: R, L and C in series and parallel circuits; Impedance; Series and parallel resonance; Transformers; A.C. meters. In Part 5 there are only three sections: D.C. generators and motors; Alternators; A.C. motors.

Seventh Symposium (International) on Combustion

London: BUTTERWORTHS SCIENTIFIC PUBLICATIONS, 88 Kingsway, W.C.2. 1959. $10\frac{1}{4} \times 6\frac{3}{4}$. 959 pp. Price £11 4s. ($28).

At the invitation of the British Section of The Combustion Institute, the Seventh Symposium (International) on Combustion was held in London and Oxford from 28th August to 3rd September 1958; the Institute of Fuel also collaborated in the organization of this Symposium. An outcome has been the publication of this 959 page volume, which is a record of the papers and discussions. Although there were more than 324 papers, it has only been possible to include about 200, because of limitations in respect of time.

It would be impossible to list all the papers within the small space available here. However, the subjects are dealt with under the following main headings: Mechanisms of combustion reactions; Spectroscopy of flames; Ionization in flames; Structure and propogation of flames; Ignition and limits of inflammability; Interaction of flames and surfaces; Turbulence in flames; Combustion in practical flowing systems; Detonation and its initiation; Special fuels; Instrumentation in combustion research. Obviously, this work is of a highly specialized nature and will be of most use to those engaged in research, but it will be a useful work of reference to designers and others who are particularly interested in combustion phenomena.

Mechanics of Road Vehicles

By W. Steeds, O.B.E., B.Sc., A.C.G.I., M.I.Mech.E.

London: ILIFFE AND SONS LIMITED, Dorset House, Stamford Street, S.E.1. 1960. $8\frac{3}{4} \times 5\frac{1}{2}$. 287 pp. Price 35s.

Although a number of books have been published on the principles of engine design, we cannot think of any earlier work representing an attempt to deal adequately with the application of the principles of mechanics to the other sectors of vehicle design and operation. The volume under review, therefore, is most useful and is bound to become accepted as a standard textbook. Many of the subjects that it covers have never before appeared in a book, although they have been dealt with from time to time in technical journals and published papers. The author scarcely needs any introduction, since his name has been associated with that of Prof. Newton in connection with the book "The Motor Vehicle", which has been well known to automobile engineers for very many years.

The subject matter has been chosen primarily for students who are working for Ordinary and Higher International Certificates or for the examinations of professional institutions or colleges of advanced technology. Its arrangement is such that only the basic principles of mechanics, simple algebra, trigonometry and calculus are required for the first chapters, while more advanced mathematics is gradually introduced as the work proceeds. Throughout the work, the author has endeavoured to present the mathematics in an easily understandable form. Although the book should prove to be most helpful to draughtsmen and junior designers of road vehicles, there is also much in it that will be of interest to established engineers.

Following an introduction covering the elementary principles of mechanics, which is virtually a short revision course, there are chapters dealing with: the forces acting on a vehicle at rest; those arising from or producing linear and angular accelerations; and the forces and torques acting in both ordinary and epicyclic gearboxes and their components. Subsequently, the mechanics of vibrations and their application to vehicle suspension systems are discussed. Braking, the mechanical properties of pneumatic tyres and the directional stability of vehicles are dealt with in three more chapters. At the end of the book there is a selection of exercises, with answers, including some from examinations in automobile engineering, set by The Institution of Mechanical Engineers.

Servicing Guide to British Motor Vehicles—Volume 5

London: TRADER PUBLISHING CO. LTD., Dorset House, Stamford Street, S.E.1. 1959. $11\frac{1}{4} \times 8\frac{3}{4}$. 312 pp. Price 67s. 6d.

In the course of the decade of vehicle production spanned by the five volumes of "Servicing Guide to British Motor Vehicles", a marked change has become apparent in the basic construction of vehicles of all types. Advanced designs at competitive prices have been made possible only by the grouping of vehicles into rationalized patterns, using the maximum number of common components. Following this trend there has come about a gradual transformation of the method of presenting servicing data. Proprietary components are now dealt with in a special section, and information on individual vehicles is concentrated upon features peculiar to the application rather than upon repetitive details of components common to a range of vehicles.

Throughout all these volumes, the same basic approach to vehicle service has been adhered to, using a logical analysis of the vehicle into major units accompanied by tables and special maintenance features. However, considerable development has taken place inside this framework, always with the aim of containing more and more information in the same space. To this end, in the articles in this volume, grouped tabular data are employed for quick reference, and exploded views of the major components are also given. The scope of the tables has been widened considerably and one page is devoted entirely to engine and electrical test and tune-up data. Because of this simple presentation of information it is believed that the contents of this volume can be comprehended by anybody interested in vehicle repair, while at the same time they are sufficiently detailed and penetrating to satisfy the needs of technicians, from the apprentice to the professional engineer.

Hydraulic Handbook: 2nd Edition

Morden: TRADE AND TECHNICAL PRESS LIMITED, Morden, Surrey. 1960. Price £5 10s.

The immediate demand for the first edition of the "Hydraulic Handbook" obliged the publishers to produce this second edition within two years of the first. In the new edition the pattern of the first is closely followed, in respect of style and presentation of material and data, but a considerable amount of new information and technical matter not previously published has been added. The volume is divided into three main sections. By far the largest, the first section occupies over half of the work. It explains the basic principles of hydraulics, and goes on to survey the various items of equipment and components, from accumulators to valves, that go to make up a hydraulic system. Then it enumerates the applications of hydraulics to industries and end-products, such as those concerned with agriculture, aircraft, automobiles, locomotives, machine tools, presses, general engineering, mechanical handling and mining.

Technical data in the form of circuits, charts and tables occupy nearly 250 pages of the second section. In this part of the book are 91 hydraulic and electro-hydraulic circuits, 126 tables with working examples, 23 nomograms, 23 charts, formulae, British Standard Specifications and American J.I.C. symbols, and it includes a hydraulic dictionary. The third and final section is a buyers' guide comprising a trade names index, a classified list of suppliers of hydraulic equipment and components and an alphabetical index of manufacturers with their addresses. The volume will be of value to designers and manufacturers of equipment incorporating hydraulics, users and buyers of hydraulically operated equipment and machinery and to consulting engineers and technical and research personnel.

Jet Propulsion Engines

London: OXFORD UNIVERSITY PRESS, Amen House, Warwick Square, E.C.4. U.S.A: PRINCETON UNIVERSITY PRESS. 1959. $10 \times 6\frac{3}{4}$. 799 pp. Price 140s.

During the past decade, rapid advances on problems associated with high-speed flight have emphasized the need for a comprehensive and competent treatment of the fundamental aspects of the aerodynamic and propulsion problems involved, together with a survey of the appropriate underlying basic sciences. The book under review is Volume XII of a series treating in sequence the following subjects: elements of the properties of gases, liquids and solids; combustion processes and chemical kinetics; fundamentals of gas dynamics; viscous phenomena; turbulence; heat transfer; theoretical methods in high-speed aerodynamics; applications to wings, bodies and complete aircraft; non-steady aerodynamics; principles of physical measurements; experimental methods in high-speed aerodynamics and combustion; aerodynamic problems of turbomachines; the combination of aerodynamic and combustion principles in combustor design; and, finally, problems of complete power plants.

In Volume XII the principles and problems encountered in combining components to form a complete engine are considered. The section headings are as follows: Historical development of jet propulsion; Basic principles of jet propulsion; The turbojet engine; The turboprop engine; The ramjet engine; Intermittent jets; The liquid propellant rocket engine; Solid propellant rockets; The ram rocket; Jet rotors; Atomic energy in jet propulsion; Future prospects of jet propulsion. This is a book of the highest standard and can be unhesitatingly recommended to professional engineers and advanced students.

The Structure of Metals: A Modern Conception

London: ILIFFE AND SONS LIMITED, Stamford Street, S.E.1. 1960. $8\frac{3}{4} \times 5\frac{1}{2}$. 118 pp. Price 25s.

Every year since 1947 The Institution of Metallurgists has held a refresher course for its members, during which four leading authorities survey the present state of knowledge in relation to a particular aspect of metallurgy. Originally, these papers were republished in book form solely for the benefit of Institution members, but the information they contained was considered far too valuable for only a limited circulation. Therefore, in conjunction with the present publishers, it was arranged that all papers read at the refresher courses from 1956 onwards should be made available to a wider public. The response to the publication of the first two volumes under this scheme was most gratifying, and the publishers feel that the demand for this third volume will certainly be at least as satisfactory.

The theme of the 1958 refresher course was a review of some of the modern theories of the structure of metals. In the first paper Professor Raynor deals with developments that have taken place over the past twenty to thirty years in the electron theory of metals, and it concisely outlines modern concepts of this highly complex subject. Dr. Catterall, in the paper that follows, describes the experimental techniques that have been developed to test the theory and to establish such things as the width of the valence band, the shape of the density of states curves, the shape of the Fermi surface, the numbers and signs of charge carriers in electrical conduction, and the change of the foregoing properties with alloying.

Professor A. G. Quarrell discusses the dislocation theory of plastic deformation, which has completely revolutionized thought about the structure and mechanical properties of metals. In the final paper, Dr. Nutting shows that this theory, at first received with some scepticism, has now been thoroughly justified by experiments, the techniques of which he describes. The contents of the book are as follows: The electron structure of metals; Experimental aspects of the electron theory of metals; Dislocations in metals; Seeing dislocations.

Work Study

By R. M. Currie, C.B.E., M.I.Mech.E., M.I.C.E., M.I.Prod.E.
London: SIR ISAAC PITMAN AND SONS LTD., Pitman House, Parker Street, Kingsway, W.C.2. 1960. $8\frac{3}{4} \times 5\frac{1}{2}$. 232 pp. Price 22s. 6d.

Experience and knowledge gained by the author, especially since 1947 in I.C.I., has been used to good advantage in the preparation of this book on work study. The volume is based on a series of three booklets in the "Outline of Work Study" series previously published by the British Institute of Management, but to expand the material contained in these booklets, extra chapters have been added. This work is intended to be basic rather than exhaustive, and, during its preparation, the author had in mind the requirements for Part II of the examination for Corporate Membership of the Institution of Mechanical Engineers, the examination of the Institution of Production Engineers, and that for entry to the Work Study Society.

However, the book is intended to appeal not only to examination students but also to other people, whether in industry or not, desiring information on this subject. It is intended primarily to provide a basis from which more specialized knowledge can be built up as required. The contents are as follows: Historical, the pioneers; Productivity and work study; The human context of work study; Method study, introduction; Method study, select and record; Method study, examine; Method study, develop and submit; Method study, install and maintain; Work measurement, introduction; Time study, principles and procedure; Relaxation and contingency allowances; Synthesis from elemental data; Analytical estimating; Activity sampling; Target times for jobs; Unoccupied time (UT) and various allowances; Confirming work content and standard times; The work specification; Work study as a service to management.

Who's Who in the Motor Industry

London: TEMPLE PRESS LIMITED, Bowling Green Lane, E.C.1. 1960. $8\frac{3}{4} \times 5\frac{1}{2}$. 592 pp. Price 42s.

The Fourth Edition of "Who's Who in the Motor Industry" incorporates many changes of appointments, company alignments, addresses and telephone numbers that have taken place in the industry since the previous edition was published. It offers an up-to-date and accurate source of information covering all sections of the industry in Great Britain. Among the new features of the latest edition is a section on the motor industry abroad, in which are listed the overseas subsidiary and associate companies of the British motor manufacturers, together with the names of their directors and chief executives. The names and addresses of competitions and racing managers are also included as a new feature, together with a complete list of one-make car clubs.

The first part of the book covers 207 pages, and deals with various aspects of the motor industry; Part II comprises 51 pages and gives the organizations and associations; in the next 14 pages is a press guide; and, finally, there are 170 pages of biographies.

Elsevier's Automobile Dictionary

Compiled by Ing. G. Schuurmans Stekhoven
London: D. VAN NOSTRAND COMPANY LTD., 358 Kensington High Street, W.14. 1960. $9\frac{1}{2} \times 6$. 946 pp. Price £7 7s.

In the basic table that comprises approximately the first three-quarters of this work there are 5,225 terms in English, each followed by the corresponding translations, in French, Italian, Spanish Portuguese, German, Russian and Japanese, ranged in columns across each double page. The English list is in alphabetical order. Following this section are alphabetical lists for each of the seven languages other than English, with numerical keys to the basic English list. Therefore, it is possible, starting from any one language, to translate rapidly into any of the other seven. Official standardized terms are clearly indicated, and divergence of usage between American and British English, European and Latin American Spanish, and European and Brazilian Portuguese, are clearly indicated. This is certainly one of the most useful technical dictionaries that it has been our pleasure to review.

Small Gas Turbines and Free Piston Engines

By A. W. Judge, A.R.C.Sc., Wh.Sc., A.M.I.Mech.E. M.S.A.E.
London: CHAPMAN AND HALL LTD., 37 Essex Street, W.C.2. 1960. $8\frac{3}{4} \times 5\frac{1}{2}$. 328 pp. Price 48s.

This book covers gas turbines ranging from the smallest models, of 30 to 40 b.h.p, up to those developing about 1,000 to 1,500 b.h.p. The emphasis, however, is on the medium output units. The whole range of engines includes the relatively simple smaller models, with a combined centrifugal compressor and axial or radial inflow turbine, the shaft of which is used for power take-off. At the other end of the range are the larger, more complex, models developing more than 750 b.h.p.

Typical examples of applications include portable and stationary power generating equipment, air compressors and centrifugal pumps, engines for light aircraft, including helicopters, units for boosting the take-off power of piston engined aircraft, and many others. The list of contents is as follows: Some historical considerations; The simple gas turbine; Some theoretical considerations; Gas turbine efficiencies; Improving small gas turbine performance; Automobile gas turbines; The free piston-turbine engine; The combustion and fuel systems; Materials for gas turbines; Some typical gas turbine applications.

New Plant and Tools

Recent Developments in Production Equipment

TO the range of automatic tube-bending machines built by Crippa Agostino, Italy, and marketed by the Addison Tool Co. Ltd., 28 Marshalsea Road, London, S.E.1, has been added a larger machine, designated Mediolanum 2. Its capacity for tubes bent cold and without filling is from 25 mm to 60 mm diameter in ferrous material and 25 mm to 80 mm in light alloy, in wall thicknesses from 0·5 mm to 3·5 mm. On the standard machine, the minimum inner radius obtainable is twice the diameter of the tube and the maximum inner radius is 200 mm, but for specific requirements somewhat smaller and greater radii respectively can be specially provided. The angles of curvature are adjustable up to 180 deg. Standard length is 8 ft, but other lengths can be provided on request.

All the controls, including the rotation of the table, are actuated by hydraulic pressure which is distributed to the various component parts by means of a hydraulic distributor controlled electro-magnetically. Power is supplied by a 10 h.p, 3-phase, 4-pole electric motor coupled to a high-pressure pump. Maximum working pressure is 60-80 atm. The machine is started up by a pedal switch, and the tube is clamped down and the rolls or slide are brought into contact automatically. A device for retracting the mandrel is synchronized with the automatic return of the machine to zero on the completion of operations and the unloading of the work. Datum stops are fitted on the graduated rotating table to obtain various angles of curvature on the same tube without removing it from the machine.

Bending dies are easily interchangeable, and dies for bending two tubes simultaneously can also be supplied. The speed of bending may be adjusted as required and the provision of automatic safety devices precludes malfunctioning of the machine and the risk of injury to the operator.

Centring and facing machine

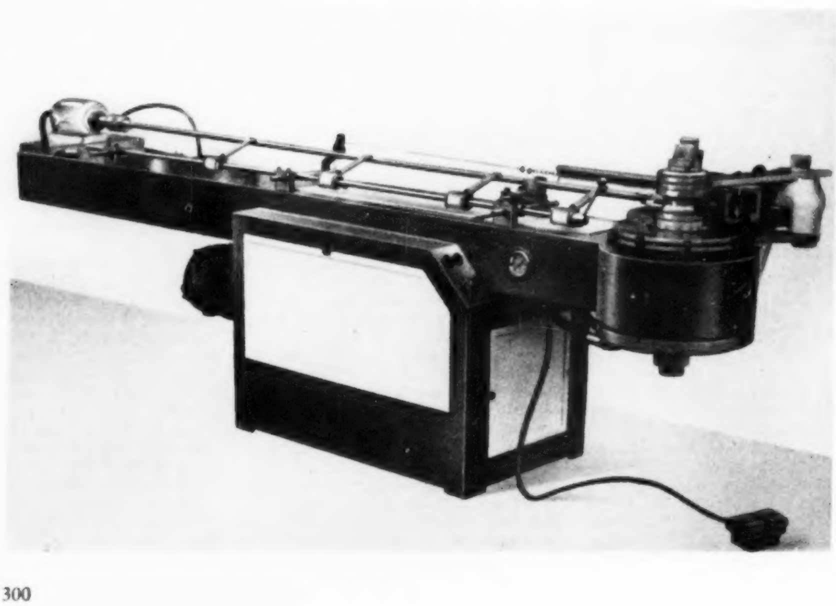
For many years centring and facing machines for use in the automobile manufacturing industry have been built by Adcock and Shipley Ltd., of Ash Street, Leicester. A recent example is a machine specially tooled and equipped to operate on a stub axle stamping in a floor-to-floor time of 45 sec. Two milling and drilling heads are mounted on a ribbed cast iron bed, with the work-holding slide mounted between them. Each head has separate milling and drilling spindles; that for milling having no feed movement while the drilling spindle has a sensitive hand feed.

Hinged sheet-metal guards are provided for all tools. Guards over milling cutters can be lifted out of the way for tool changing, but those over the centre drills are automatically operated so that complete protection is ensured until the spindles are fed inwards by the hand levers. Then the guards are lifted to allow the drills to pass beneath. In the illustration, the guards are both shown in the operating position. The heads are adjustable on the bed to accommodate variation of component length by 14 in from the minimum.

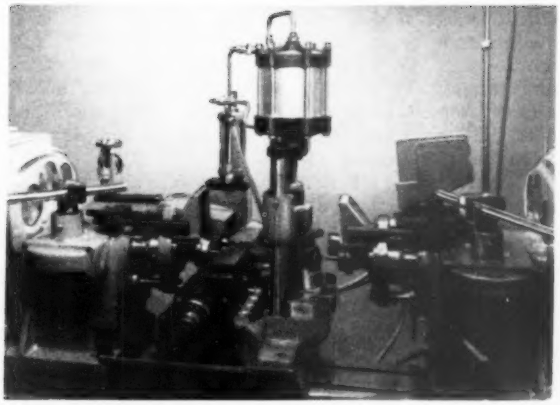
The operational sequence is as follows:

1. Load component into fixture and clamp
2. Advance work cross slide by handwheel at rapid approach rate
3. Engage automatic feed, which trips out when the milling operation is completed
4. Return cross slide by handwheel to a dead stop at the front position
5. Drill centres, using hand levers
6. Release fixture clamp and unload component.

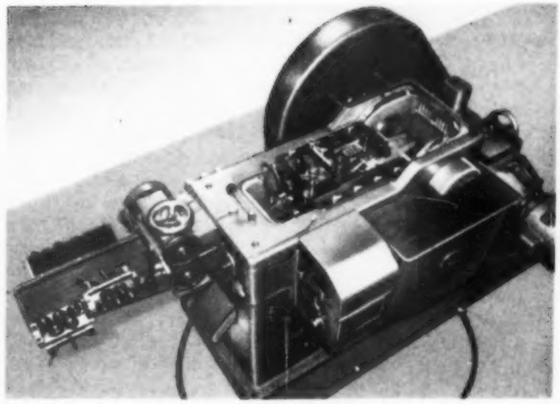
Clamping or unclamping is effected by a control valve positioned to the right of the handwheel. This valve passes



Automatic tube-bending machine
(Addison Tool Co. Ltd.)



Centring and facing machine
(Adcock and Shipley Ltd.)



Peltzer and Ehlers double-blow, cold-heading machines
(Rockwell Machine Tool Co. Ltd.)

air to two cylinders. One acts directly to clamp the work against a suitably shaped location block and the other, through a rack and pinion gearing, turns a shaft with right-hand and left-hand threads to give a self-centring action to the jaws of a vice that clamps the stem of the component.

Coolant is supplied by an electric motor-driven pump mounted in the tank at the rear of the machine. Also located at the rear, at an accessible height, is the panel on which all the electrical equipment and connectors are mounted. By the provision of suitable work-holding fixtures, a wide variety of components can be handled on these machines.

Portable hacksaw machine

Designed for use where its small size and portability are of advantage, this Braunstone hacksaw machine weighs 280 lb, occupies floor space only approximately 2 ft 6 in × 1 ft 6 in, and does not need to be bolted down to the floor. A $\frac{1}{3}$ h.p, 3-phase, 50 c/s motor drives a 12 in heavy-duty blade at 150 stroke/min and the capacity is 4 in diameter at 90 deg.

In operation it cuts on the back stroke. Hydraulic relief is provided on the forward stroke, to increase cutting efficiency particularly when cutting thin tubes or sections. The relief system is adjustable by means of a knurled screw on the oil tank. It is unscrewed about two turns to rapidly lower the overarm and returned to its original position for cutting. Should it be left unscrewed the dashpot will empty, but it can be immediately re-charged by a single raising of the overarm.

The saw frame, fitted with Oilite bushes, runs on two guide rods on the overarm. Drive is by vee-belts, giving a double reduction, and both the motor and the counter-shaft bracket are adjustable on slide rails. A suds pump and tank can be fitted if required. The machines are marketed by A. A. Jones and Shipman Ltd., Narborough Road South, Leicester.

Double-blow, cold-heading machines

The application of single-blow heading machines is limited by their maximum upsetting capacity. Without incurring the risk of buckling, the greatest length of stock that can be upset by a single blow is 2·5 times its diameter. Heads having a volume larger than 2·5 times the cross-sectional area of the wire must be produced by two blows in a double-blow heading machine. Maschinenfabrik Peltzer and Ehlers, Krefeld, West Germany, have recently introduced a series of six high-speed, double-blow, cold-heading machines to handle material from $\frac{1}{16}$ in to $\frac{3}{8}$ in diameter. The machine illustrated is the Model DKP/S6 which will produce bolts or similar components with a maximum shank diameter of $\frac{1}{4}$ in by 1 in long. Output is at rates up to 200 pieces per minute.

This machine is driven by a 7·5 h.p. motor mounted on the machine base. The motor is coupled directly to a variable-speed drive which provides steplessly variable range from 320 to 1,425 rev/min. Multiple vee belts transmit power to the large-diameter, heavy flywheel which is equipped with an effective foot brake for quickly stopping the machine after switching off the motor.

The material is taken from the coil through the adjustable rollers of the straightening device by power-driven feed rolls which are provided with a quick-acting pressure release. After being cut off to the required length, the blank is transferred to the heading position, where the preforming punch pushes it into the die before exerting the full force of its

Braunstone power hacksaw
(A. A. Jones and Shipman Ltd.)

blow. Simultaneously, the whole or part of the shank can be extruded to the diameter required. During the next revolution of the crankshaft, the finishing punch moves into position in front of the die and completes the heading operation by a single blow. The blank is then ejected from the die, an air jet assisting its downward travel to the chute from which it falls clear of the machine and into a bin.

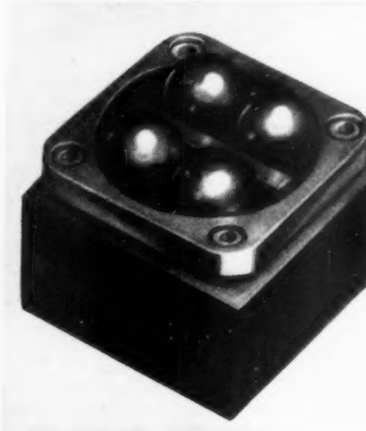
A pressure lubrication system delivers oil to all the necessary points on the machine and surplus oil drains to a large tray contained within the machine base. The position of the punch block carrying the preforming and finishing punches is easily and quickly adjusted, and resetting of the machine for different wire diameters can be effected within a very short time. An inching button is included in the electrical controls to facilitate setting and adjustment. Rockwell Machine Tool Co. Ltd., Welsh Harp, Edgware Road, London, N.W.2, are the sole selling agents for these Peltzer and Ehlers machines in the United Kingdom.

Power tool support

More complex and heavier portable power tools are finding application on both production and assembly flow lines, and the increase in efficiency made possible can be fully realized only if their manoeuvrability is maintained without increasing stress and strain on the operators. The universal tool support manufactured by Vokes Genspring Ltd., Henley Park, Nr. Guildford, Surrey, was developed to provide this facility for such equipment as welding guns, multiple nut and stud runners, riveting tools, and grinding tools.

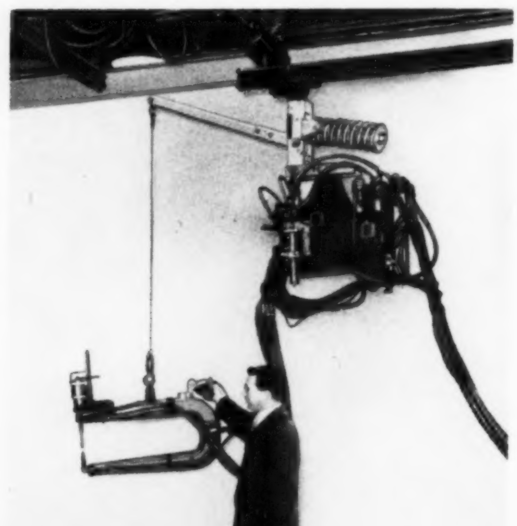
Of simple design, the support needs no maintenance since there are few moving parts and all bearings are of the self-lubricating type. Angular movement is through 75 deg and thus a relatively short lever arm sweeps through a useful length of arc to give a substantial vertical deflection. The supporting springs are mounted horizontally to minimize the headroom occupied. With the standard 40 in lever arm a deflection of 48 in is obtained and the support can meet requirements for any load from 150 lb to 15 lb. This is accomplished in three phases by altering the spring arrangement. In the first phase two concentric helical springs are used; in the second the outer spring; and in the third, the inner spring. In each case the specific load is set by means of a screwed adjuster.

The outstanding feature of the support is its lightness of action. It can be adjusted to bring a tool to rest at any specific height required or to raise it to the uppermost position when it is released. In the application shown, carrying spotwelding equipment, the support effort is variable since when the gun is elevated the heavy cables impose a greater load. For convenience in varied welding operations, two supports can be suspended from a common carrier. The operator then has two different guns at his disposal by swivelling the supports through 180 deg.



Ball-type support blocks
(Rubert and Co. Ltd.)

Universal support for power tools
(Vokes Genspring Ltd.)



Support blocks

While vee blocks commercially produced to ordinary workshop standards are relatively cheap, precision blocks made to an accuracy of ±0·0001 in must be specially selected and matched. So many factors are involved—squareness, flatness, angle of vee, position of vee, surface finish—that their cost is high. The finish of the contact faces must be exceptionally good to ensure that a component is not marked when it is revolved in the vee.

Making use of the inherent accuracy of standard steel balls, Rubert and Co. Ltd., Demmings Road, Councillor Lane, Cheadle, Cheshire, have developed a support block that requires precise machining on only two parallel surfaces, the top and the bottom of the base, and consequently can be produced at a much more competitive price. The component is supported on four balls positively located on the top surface of the block by a slightly tapered retaining ring. Pairs of balls are separated by a spacer bar, giving in effect two different angles of vee in a single block which are selected by turning the block through 90 deg. This feature, of particular value when checking for lobing, enables a single pair of standard blocks to accommodate work from 0·0156 in to 10·0 in diameter.

As the balls are hardened and polished they are easily cleaned and are not likely to retain foreign matter that would mark a component. When flat-sided components are supported there is less likelihood of matter being trapped on four ball contacts than on a plane surface. Should the balls eventually wear, as the result of checking

long runs of a particular component, the retaining ring can be slackened off and the balls turned to bring new points of contact into position. In the case of very heavy components which might suffer indentation by the balls, accurately machined flat shims are used to spread the load. For use with parts of very soft metals, nylon balls can be substituted for steel balls.

When used to support a workpiece during drilling, milling or jig-boring, a Rubert B-block, as it is termed, can be used to provide accurate indexing at 90 deg. Special blocks with more than four balls can be produced for indexing at other angles or multiples of angles.

Small precision lathe

A universal instrument and toolmakers' lathe, with a comprehensive range of attachments and equipment is now produced by the Ideal Machine Tool and Engineering Co. Ltd., of Kingsland Road, London, E.8. It is made to the same high standards of precision as the maker's well-established "Ime" watchmakers' lathe. Beds of two different lengths are available, giving a work capacity between centres of 12 in and 16 in. Centre height above the bed is 3 in and the maximum collet diameter is 0·5 in.

Of tool steel, precision ground and hard scraped, the v-shaped bed is mounted on two cast iron box supports. The support at the headstock end accommodates the driving motor reversing switch. Headstock and tailstock are ground on a master bed to ensure accurate alignment. In the solid steel headstock the nickel-chrome steel spindle runs in high-carbon steel conical bearings which are adjustable by means of a special nut. Both spindle and bearings are hardened, ground and lapped. Lubrication is by large, spring-loaded grease cups and the bearings are protected by dust covers and felt washers.

On the face of the largest diameter of the three-diameter driving pulley 60 holes are provided for use in conjunction with an indexing attachment for dividing purposes. American-type collets with external threads are used in the spindle and in the tailstock and are closed by draw-in bars operated by a large plastics knob. A collet closer can be fitted at the headstock and the tailstock sleeve can be equipped with a lever-type feed for drilling operations. An adjustable depth bar, graduated to 0·04 in is incorporated to set up depth of drilling.

Each lathe is tested at a speed of 5,000 rev/min for eight hours, dismantled, cleaned, reassembled, and finally rigorously inspected. A test chart is issued with each tool.

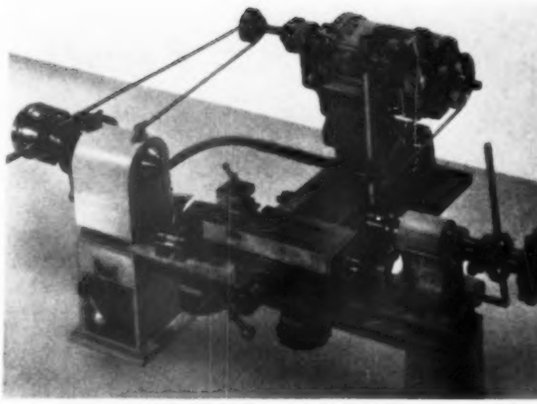
Automatic punch presses

Credited with high-speed performance and extreme accuracy, Swiss-built Bruderer precision punching presses in three sizes of 6-, 12-, and 20-ton capacity respectively are now marketed in Britain by the Press and Shear Machinery Co. Ltd., 172-178 Victoria Road, Acton, London, W.3. Basically, the design stems from that for a three-pillar die-set. The guide pillars are located, two at the front and one at the rear, at the points of an equilateral triangle, at the geometric centre of which is the thrust-point of the slide. Pillars are tempered, ground, and lapped, and the oil-well type slide bushes are of super-finished phosphor bronze. All the main bearings of rotary parts of the press are either single-row or double-row needle-roller bearings.

Accuracy in the main bearings is a contributory factor to the consistent precision of the feeding device, the transport rollers of which are also mounted on roller bearings. For a short feed up to about 1 in an accuracy of 0·0005 in can be attained, while on long feeds up to the maximum of 6¾ in an accuracy of approximately 0·005 in can be expected. Feed length is steplessly variable and the normal range of strip widths for the three models is 1¼ in, 2½ in and 5½ in, with thickness up to 10 S.W.G. Strip passage width between the pillars is about 6¾ in, with a maximum of 4 in if symmetrical about the ram axis. Ram stroke is variable up to 1¼ in, the position of the slide on the ram is adjustable over 2 in, and the drive can be either in steps or steplessly variable up to 1,500 strokes/min.

The excellent alignment maintained by the triple-pillar guide system is conducive to long working life of the press tools. Runs of 2½ and 3 million components are commonly obtained before a regrind becomes necessary.

A range of auxiliary equipment is available, including strip coil reels, winding-off reels, strippers, lubricators, and five-roll straighteners. In addition to standard strip feeds, special details for wires and profiles can be supplied.



Instrument and toolmakers' lathe
(Ideal Machine Tool and Engineering Co. Ltd.)

Bruderer precision punching press
(Press and Shear Machinery Co. Ltd.)

Cooling Fans

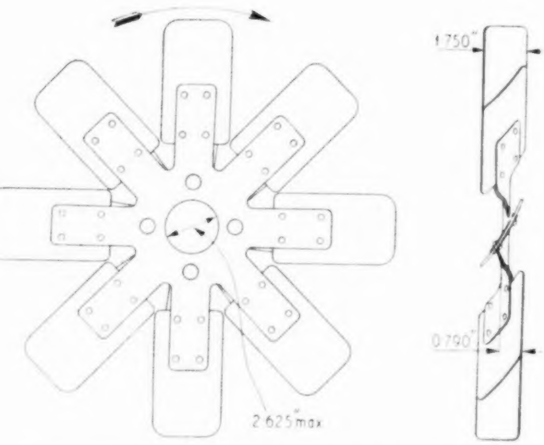
Wilmot Breeden Range of Types and Sizes Adaptable to Meet the Requirements of all Vehicle Engine Cooling Installations

PROPELLER-TYPE fans in a varied range of patterns and sizes have been manufactured by Wilmot Breeden Ltd. for many years. In the smaller diameters their application is as radiator fans for petrol and diesel engines in both mobile and stationary service. Larger components are in demand for ventilation, cooling, heating, air circulation, and smoke or fume extraction. Technical resources, production capacity, and a skilled team of designers are grouped at the company's new plant at Colley Lane, Bridgwater, Somerset. Recently, an extensive programme of development has been undertaken and production has been reorganized. As a result, ranges of fans of various types and of proved performance have been standardized to meet all normal requirements in the fields covered. Problems can be investigated and special designs prepared when specifically required.

One design series of fans is suitable for ducted installation in the coolant systems of diesel engines. Their respective delivery capacities cover the requirements of commercial and passenger vehicle power units up to those for relatively large stationary generating sets. All are of a fabricated mild steel sheet construction. A pair of spider pressings spaced by an annular stiffening ring forms the hub and the separate blades, sandwiched between the arms of the spiders, are riveted in position. To achieve a measure of rationalization, each of the four sizes of spiders can accept a range of different blade lengths to give a total of five six-blade and fourteen eight-blade fans of from 15 in to 28 in diameter in increments of one inch.

Hubs can be bored, up to a designed maximum diameter, and the number and size of fixing bolt holes arranged to suit specific requirements. In every case the spider pressings are in 16 S.W.G. material, but blades and stiffening rings are in 18 S.W.G. in fans up to 19 in diameter and in 16 S.W.G. for fan diameters from 20 in to 28 in. The weight range is from 4 lb to 11·5 lb.

Standard fans are built for clockwise rotation and are symmetrical about the hub so that they can be inverted on installation to reverse the direction of air flow. Designed maximum speeds of rotation are from 4,500 rev/min down to 3,000 rev/min for fans from 15 in to 28 in diameter. All are intended for ducted installation, and over the same range of diameters the recommended tip clearance varies from 0·16 in to 0·24 in.



This 15 in fabricated fan is typical of a range from 10 in to 40 in diameter. Dimensions relate to the 15 in to 19 in group

Control room at the wind-tunnel test house

Additional to the main range there are smaller and larger series of fans to the basic design; 10 in to 14 in diameter and 29 in to 40 in diameter respectively. Variants of the standard design are designated as lightweight and as heavy-duty fans. Lightweight units have mild steel spiders fitted with blades of 16 S.W.G. aluminium alloy. In the case of a 24 in diameter fan, weight is reduced from 10 lb to 6 lb 10 oz, but blade angle is smaller and pressure, air volume, and speed characteristics are all lower than for the standard version. Heavy-duty fans have a greater blade angle than standard and are designed for conditions requiring a large-volume air flow. Here, again, steel spiders and aluminium alloy blades are used, but the spiders are of a different pattern to provide a large diameter bore. A spigot ring is riveted to the hub assembly for reinforcement. Taking a 24 in diameter fan again for example, weight is down from the standard 10 lb to 7 lb 6 oz.

The line drawing of the standard eight-blade fan of 15 in diameter is typical of the complete range, and the performance curves also relate to this particular unit. All curves have been corrected to standard air; that is, to air having a density of 0·0764 lb/ft^3 at 60 deg F and 30 in Hg barometric pressure. Similar performance charts are available for each standard unit and intermediate values or performance under different conditions may be evaluated from the following equations:

Providing the air density remains constant—

Volume (Q) varies directly as the fan speed (N) $$Q_2 = Q_1 \left\{\frac{N_2}{N_1}\right\} \qquad (1)$$

Pressure (P) varies as the square of the fan speed $$P_2 = P_1 \left\{\frac{N_2}{N_1}\right\}^2 \qquad (2)$$

Power absorbed (HP) varies as the cube of the fan speed $$HP_2 = HP_1 \left\{\frac{N_2}{N_1}\right\}^3 \quad \ldots\ldots\ldots\ldots (3)$$

Air density varies with temperature, $d = \frac{1{\cdot}5Pa}{T}$ where Pa = absolute pressure in lb/in^2 (Pressure gauge + pressure barometric) and T = absolute temperature (deg C + 273 deg). If the air density (d) is varied and the speed and size (D) remain constant, then the volume of air remains constant.

Pressure varies directly as the relative density $$P_2 = P_1 \left\{\frac{d_2}{d_1}\right\} \quad \ldots\ldots\ldots\ldots (4)$$

Power absorbed varies directly as the relative density $$HP_2 = HP_1 \left\{\frac{d_2}{d_1}\right\} \quad \ldots\ldots\ldots\ldots (5)$$

When the fan diameter is varied, while speed and air density remain constant—

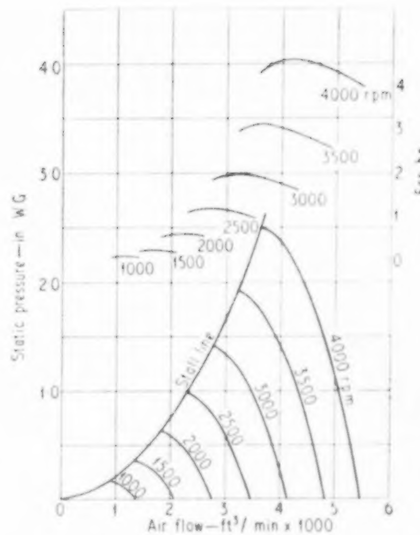
Volume varies as the cube of the fan diameters $$Q_2 = Q_1 \left\{\frac{D_2}{D_1}\right\}^3 \quad \ldots\ldots\ldots\ldots (6)$$

Pressure varies as the square of the fan diameters $$P_2 = P_1 \left\{\frac{D_2}{D_1}\right\}^2 \quad \ldots\ldots\ldots\ldots (7)$$

Power absorbed varies as the fifth power of the fan diameter $$HP_2 = HP_1 \left\{\frac{D_2}{D_1}\right\}^5 \quad \ldots\ldots\ldots\ldots (8)$$

Obviously, equations (6), (7), and (8) can be applied only to fans that are geometrically similar. For instance, the performance characteristics obtained from a fan having the spiders for the 15 in-19 in diameter group cannot be used to predict the performance of a fan using the spiders for the 20 in-23 in diameter group, nor for a fan having blades at a different angle.

The performance of each fan of standardized diameter in each type is evaluated in a wind tunnel arranged with readily interchangeable throats, intake sections and discharge sections. This test equipment is fully instrumented and remotely controlled from a room giving a full view of the fan under investigation and the driving motor. A chain mesh curtain suspended outside the control room window safeguards the test engineer without unduly obscuring the view. Drive to the fan shaft, through multiple vee belts, is steplessly variable and rotational speeds are indicated by a stroboscopic device. A comprehensive range of tests is



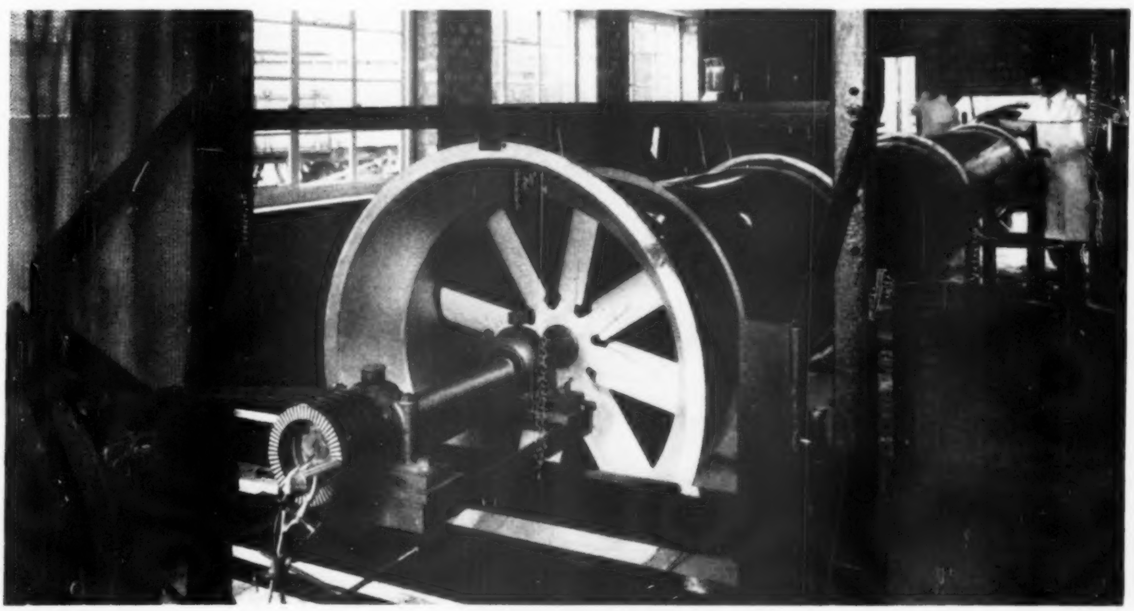
Performance curves for 15 in diameter, eight-blade fan. All fans have an accurately calibrated performance, determined by wind-tunnel tests

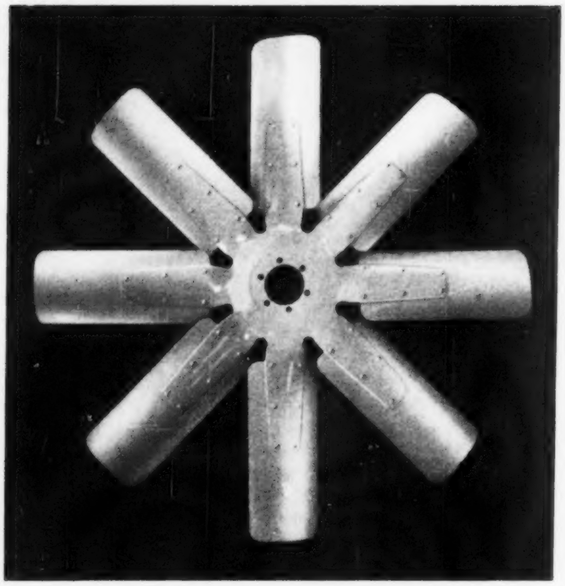
conducted in each case, so that each standardized fan has an accurately calibrated performance chart on record.

New designs in course of development are thoroughly investigated in this test house before being accepted for standardization and production. As a service to customers, fans of their own design can be tested to assess performance, suggest improvements, or recommend change to a proved design. Problems of excessive power absorption and noise can be studied and solutions sought. General problems of installation, radiator size, shape, and capacity can be investigated in conjunction with specific fans.

Problems of noise are frequently encountered in vehicle installations, and may be of some consequence in so-called quality cars. Objectionable noise levels commonly arise from harmonic vibration excited at certain engine speeds

The wind tunnel at the Bridgwater plant, where all standard fans are calibrated, developments are tested, and problems investigated





Eight-blade, 33 in diameter fan, representative of the larger series for stationary diesel engine installations and ventilating systems

by a fan having a certain number of evenly pitched blades working in some proximity to a stay or structural member on the chassis or bodywork. To circumvent such difficulties the firm has developed a number of fans, of different types, with the blade arrangement unequally pitched in order to break the harmonies. Two such five-blade, asymmetrical fans are illustrated, one of the fabricated spider-hub type and one of the simpler pressed type. It is, of course, not invariably necessary in such cases to resort to an asymmetrical design. The removal of two blades from a standard six-blade or eight-blade fan of suitable diameter is often sufficient to reduce noise to an acceptable level.

Cooling fans for the engines of cars in high production must be of light construction and necessarily of competitively low cost. Wilmot Breeden manufacture two-, four-, five-, and six-blade fans pressed from mild steel strip or plate in a wide range of diameters. Considerable ingenuity is displayed in obtaining the requisite stiffness in the pressings across the hub and blade root sections. Usually the blades are offset from the plane of the hub portion, which has no central bore but instead is pressed up into a fairly deep central boss. From this boss, tapering ribs extend across the hub, over the transition area, and finally run out on the blade portion. The construction admirably achieves its purposes but imposes a slight limitation on the arrangement of fixing bolt holes, necessitating a four-bolt fixing for two- and four-blade fans and a three-bolt layout for six-bladers.

Examples of fans manufactured from strip material are shown in two- and four-blade designs. In the last mentioned unit the hub of the overlying two-blade member is formed with two additional ribs which nest over the ribs of the member below. Fans pressed from a single sheet of material are shown in six-blade symmetrical and four-blade unevenly pitched versions. The five-blade asymmetrical model is built up of a two-blade and a three-blade member nested together at the hub. To facilitate assembly and fixing in this instance the hub central boss is of much greater diameter and in it are provided the four fixing holes.

It is the policy of Wilmot Breeden Ltd. to develop and manufacture ranges of fans of different types, all of sound design and proved performance, to enable a selection to be made to meet requirements on a scientific rather than on an empirical basis. Complementary to this policy is the provision of facilities for the investigation of power absorption, noise, and installation problems, and also the development of fans of special design for specific requirements.

Examples of the wide variety of standardized fans for vehicle engines. Left, four- and six-blade single-pressing type. Centre, fans produced from strip material. Right, five-blade asymmetrical fans in fabricated and pressed sheet versions

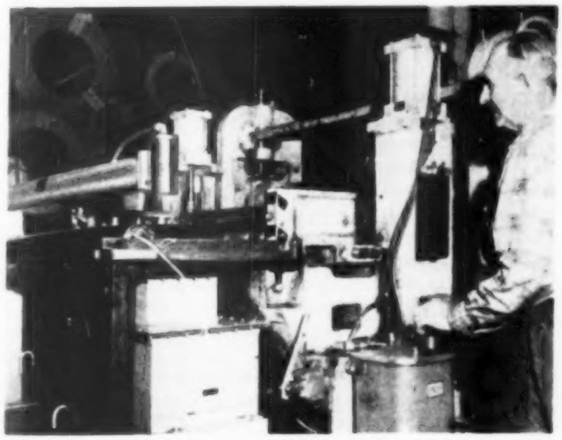
Gauging Coiled Strip Material

Mobile X-ray Thickness Gauge for Strips up to 54 in Wide on Slitting or Shearing Lines

In rolling mill operations, non-contacting X-ray gauging equipment is used to measure the thickness of the steel or aluminium material being processed at speeds up to 6,000 ft/min. Similar equipment is now available for incorporation in slitting and shearing lines for checking stock or warehoused coiled material before being passed into production. By the elimination of out-of-tolerance strip and rejects due to faulty material, it is claimed that production has in some instances been improved by as much as 30 per cent.

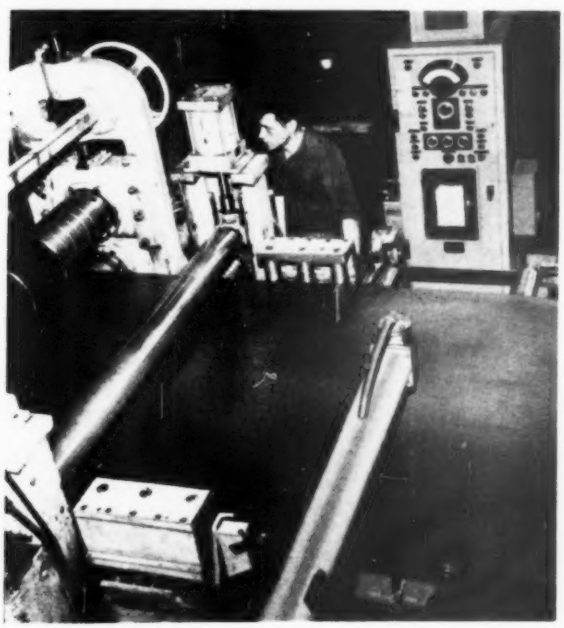
Thickness measurement of coiled strip in widths up to 54 in are made by passing a pencil-sized X-ray beam through the material and collecting the transmitted energy in a receiver unit. All measurements are continuous, from side to side and end to end of the strip, and can be taken at high speeds up to the rate mentioned earlier. Energy collected in the receiver is converted into an electric signal which is used to display the thickness measurement on a dial and record it on a moving chart installed in a supervision console sited conveniently alongside the slitting machine. Known as the XactRay, this gauging equipment was developed and is manufactured by the Weston Instrument Division of Daystrom Inc, of Murray Hill, New Jersey, U.S.A, and is marketed by the International Operations Group of Daystrom Inc, at Newark, New Jersey.

In practice, the thickness specifications of strip material may range from the usual commercial tolerance of approximately plus or minus 10 per cent to tolerances as close as plus or minus a few per cent. The gauging equipment, therefore, has to be capable of indicating accurately, on an easily readable scale, thickness variations of as much as 0·005 in or as little as 0·00004 in. During slitting operations a strip record of thickness is made of all material. Each coil can be given a mean reading or, if required, the graph can be released with the coil to the production department. By this means, it can be readily ascertained that the coil, or some part of it, conforms to the stipulated tolerance before work is commenced.



Operator adjusts power source for X-ray gauging equipment installed on a Ween slitting machine

XactRay gauge receiver unit traverses the wide strip as it is run through the slitting machine



Previous practice was to take spot checks with a hand micrometer during processing. This was time-wasting and also largely ineffective, since a micrometer could check only a limited distance from each margin of the material. Rolling mill processes commonly produce a slight crown effect on the coil and, thus, there was no adequate method of determining the thickness of the mid-section until the strip was slit or cut. In addition to its accuracy and speed of measurement, an obvious advantage of the XactRay gauge is the ability to scan and record the thickness over the entire width and length of the strip before slitting or cutting.

The gauge measures and records the nominal thickness in multiples of 0·001 in and that value is centred on the meter. Plus or minus deviations from the nominal value, in increments of 0·001 in, are indicated continuously on the meter and recorded on the moving chart. If significant portions of the strip are outside tolerance, automatic alarm signals are actuated, the machine is stopped and, if necessary, the coil is unloaded and another substituted.

All setting of the gauging equipment is carried out by the machine operator. By simple dial adjustments he sets the gauge to the specified nominal thickness and to the tolerance limits and then begins slitting operations. Since no physical contacts are made and there are no moving parts in the gauging head there is virtually no wear. The X-ray tube is energized at so low a level that it lasts indefinitely without measurable deterioration, and maintenance of the equipment consists of an occasional electrical check-up.

The illustrations show the XactRay gauge installed on a Ween slitting machine in one of the stock-holding plants of the Coil Steel Corporation, of Cleveland, Ohio, U.S.A.

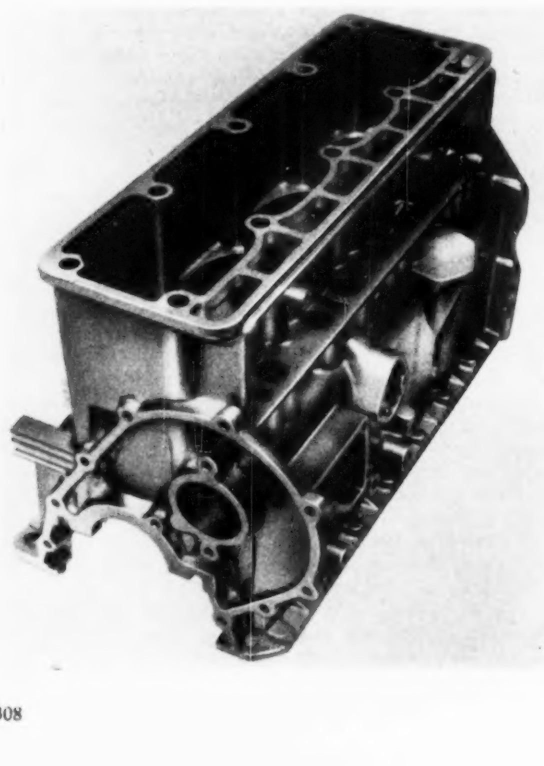
LARGE LIGHT-ALLOY DIECASTINGS

New Triulzi Machines of 1,500-ton and 2,200-ton Clamping Pressure Respectively, Built to Russian Order, for the Production of Automobile Components

SINCE the early years of the industry, the use of aluminium has been advocated for major structural components of engine and transmission units on account of its lower specific weight when compared with grey cast iron, the conventionally used material. Although its suitability for such duties was amply demonstrated in the rapid expansion of the aircraft engine industry during World War I, it made but little impact on the automobile industry in subsequent years. Many factors were responsible for the slow progress made by the attractive, lighter metal. It was more costly; less was known, in design office, foundry and machine shop, of its physical and mechanical properties; an indigenous supply of bauxite, the basic raw material, was lacking, as was water power for the economic generation of the electricity required for the conversion of bauxite; there was concern about the adequacy of supplies of the metal and also of the possibility of interruption of supplies; and compensatory advantages in respect of casting and machining processes were insufficient. One manufacturer of aluminium wheels for lorries advertised at that time in the *Automobile Engineer* to the effect that the higher initial cost was partly offset, at the end of the useful life of the component, by the higher scrap value of aluminium relative to that of cast steel.

Some of these factors have been eradicated and others are of lessened import. Higher cost, however, still remains, and is unlikely to be materially reduced in the foreseeable future since more energy is consumed specifically in the conversion of aluminium than of cast iron. The advent of mass production methods scarcely influenced the issue. If anything, it tended to increase the disparity between the costs of aluminium and iron as, before World War II, the cost of material stood higher in relation to the cost of labour.

Produced on the Triulzi 1,500-ton clamping pressure machine, this bell housing casting in Alpax weighs 29·5 lb



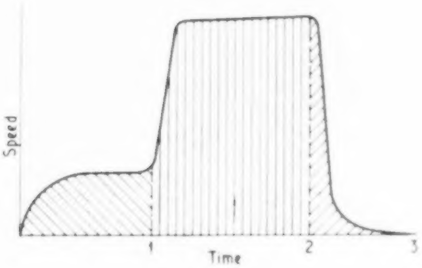
Light alloy engine block for 2,446 cm³ automobile engine. Production rate is not less than twenty castings per hour

Today, flow-production methods, transfer machining lines, and automation have radically changed the situation. Time has assumed prime importance and aluminium makes possible faster cycle times both in the foundry and on the machining line.

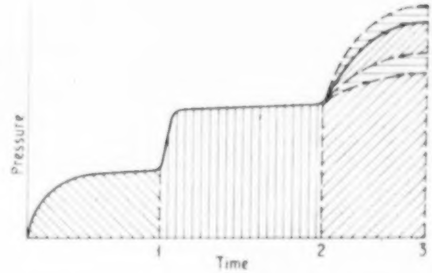
Gravity or low-pressure, permanent-mould castings or pressure diecastings can be consistently held to closer limits than sand castings. Machining allowances can be reduced and the need for roughing cuts disappears. Fettling is reduced to the simple cut-off of sprues. As regards machining, cutting speeds are higher for aluminium than for cast iron. Machine tools can be lighter and faster running, occupy less space and require less power. Single cuts at faster cutting speeds can so reduce time on such operations as face milling that the need to duplicate machines in order to obtain a reasonable cycle time for a transfer machine line is obviated. As has already been reported,* a changeover from cast iron to aluminium presents the opportunity to shorten the length of a transfer machine line, saving valuable shop space and lessening power requirements.

These advantages, which can adequately compensate for the higher cost of the metal, can best be realized by the pressure diecasting process. It has been used with marked success for motorcycle, scooter and outboard motor components but to date has found application only for minor components on automobiles. The General Motors Corvair

* *Automobile Engineer*, February, 1960.



The phased casting process. Left: speed-time diagram of ram movement. Right: variation of pressure on a time basis



air-cooled cylinder block is a low-pressure casting. Apart from the general considerations, the main deterrent has been the size of motor car components. As the projected area of components is increased, so the total pressure required to close the die parts rises and for items of the size of a four-cylinder or six-cylinder engine block truly massive machines are required. Such machines were not available in Europe, although one had been built in the U.S.A.

A description of the range of diecasting machines manufactured by the Italian firm of A. Triulzi, of Milan, has been given in an earlier issue.† This included the largest machine, then newly introduced, which has a clamping pressure of 1,000 tons. About a year ago the firm received an order from Soviet Russia for machines of 1,500-ton and 2,200-ton clamping pressure and also for the dies to produce on the respective machines a large bell housing for a tractor vehicle and a four-cylinder, water-cooled block casting for an automobile engine. In general, the machines follow the basic design of the standard 1,000-ton model, being of the cold-chamber, direct-locking, water-hydraulic type and provided with automatic metering and ladling equipment. Dies are made by another Italian firm, Fonderpress Di Gamberini Tagliavini, of Bologna, specialists in such work and long associated with the Triulzi concern. The smaller of the two machines and its dies has already been proved and dispatched to Russia; the larger machine is approaching completion and will be delivered this autumn.

The example of the bell housing casting illustrated was one of the test specimens produced in the Triulzi works. It is cast in Alpax (LM6) and the as-cast weight, with sprue and slug, is 29·5 lb, while the trimmed weight is 24 lb. The projected area of the casting is 342 in^2 and the ram diameter is 5·125 in. An injection pressure of 84 tons is used, giving a pressure on the metal of approximately 9,000 lb/in^2. The general section thickness of the casting is about 0·375 in. Production is at the rate of 30 shots per hour.

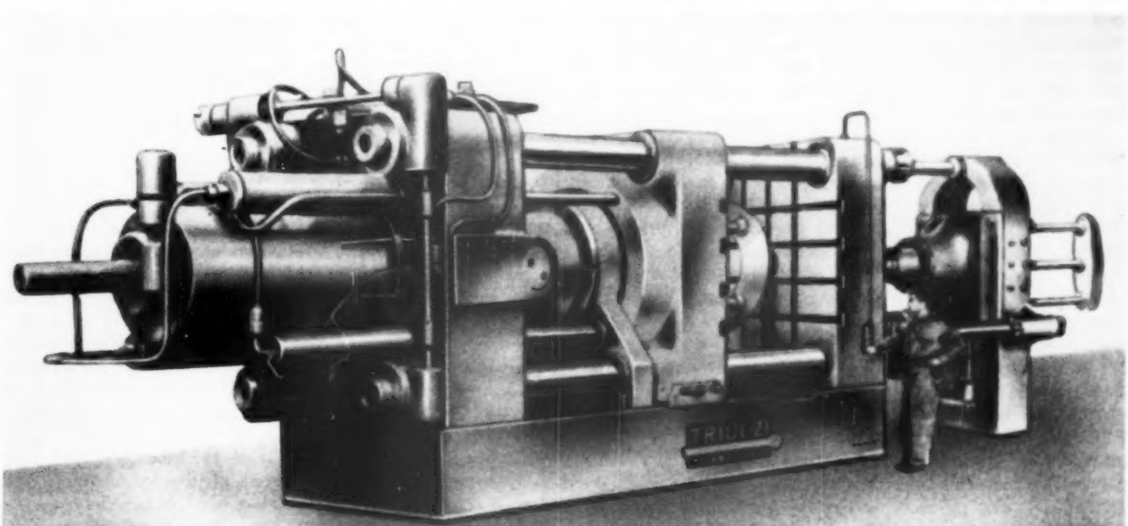
Overall dimensions of the machine, excluding the hydraulic pump and motor and a 220-gal accumulator, are 28 ft 4 in × 5 ft × 8 ft 9 in and the weight is about 50 tons. Water consumption is 16 gal per shot. Dimensions of the platens are 58·7 in wide and 66 in high, with a useful height between the machine tie bars of 44 in. Maximum daylight between the platens is 65 in and the stroke of the moving platen is 38·5 in. Normal working pressure is 2,130 lb/in^2 and the pressure on the metal can be from that value up to 28,400 lb/in^2.

It is reported that the four-cylinder engine block is for a 2·5 litre automobile engine. Presumably, this refers to the engine of the Volga car which in its current version with a cast iron block has a bore and stroke of 92 mm × 92 mm giving a swept volume of 2,446 cm^3. This unit has a compression ratio of 7·5 : 1, push-rod operated overhead valves, and a single downdraught carburettor. It has a maximum torque of 130 lb-ft and develops 80 h.p. at 4,000 r.p.m.

A silicon alloy—reputedly 5 per cent silicon and 0·5 per cent copper—is used for the engine block diecasting, which has a gross weight of 50·6 lb including sprue and slug.

† *Automobile Engineer*, April, 1955.

The 2,200-ton clamping pressure machine, with automatic metering and ladling equipment, can handle aluminium charges of up to 99 lb



When trimmed, the weight of the casting is reduced to 41·8 lb. It is of the open type to accommodate detachable inset liners, as in the original cast iron version. The construction is stiffened by the bosses for the holding-down bolts, these bosses being continued down to the seatings of the main bearings, and is braced by the light alloy cylinder head casting. Despite its weight, the block casting does not tax the capacity of the machine which can handle a charge of metal up to 99 lb in weight. The rate of production will, it is understood, be not less than 20 of these castings per hour.

Details of the machine are impressive. Overall dimensions of the machine alone are 49 ft × 7 ft × 13 ft high and its weight is approximately 100 tons. The hydraulic pump, driven by a 55 h.p. motor, delivers 27 gal/min and the recommended accumulator has a capacity of 666 gal. Consumption of water is 27 gal per shot. Platen size is 72·8 in wide by 76·7 in high; the clear height between the tie bars is 51·5 in. Working pressure is 2,130 lb/in^2, as on the smaller machine, and the injection force can be from 89 tons up to 225 tons. The force available for ejection is 35 tons. Minimum and maximum thicknesses of dies are 27·5 in and 47·25 in respectively and the moving platen has a stroke of 59 in.

On these large machines the casting process is phased in three stages to ensure the production of sound castings of even density. This phasing as it affects ramming speed and pressure on a time basis is shown in dimensionless graphs. On the speed-time graph the period from 0 to 1 indicates the initial slow movement of the ram which allows the evacuation of air from the injection sleeve. In the period from 1 to 2 the ram attains a higher speed, depending upon the characteristics of the casting. When point 2 is reached, the die cavity is virtually filled and thereafter, between points 2 and 3, the ram speed is decreased in order to obviate the production of flash.

The pressure-time graph shows from 0 to 1 that the material is pressed with a specific force into the cavity of the die. In the second phase from point 1 to 2 the value of the preceding pressure is raised as the ram is advanced. When the cavity has been completely filled, at point 2, a higher final pressure is applied for the period from point 2 to point 3; this increased pressure improves the density of the casting and its mechanical properties.

Post-Graduate Course in Automobile Engineering

AN Advanced School of Automobile Engineering, in association with The College of Aeronautics, is being established at Cranfield, Bucks. The purpose of the school is to provide a course of studies in the application of science and technology in automobile engineering. As a result of continuous development, techniques of design, research and production of road vehicles have reached a very advanced stage; so, in order to obtain further improvements, a sound knowledge of scientific and engineering theories, together with an understanding of the economic and production problems involved, is required. At Cranfield the young automobile engineer will benefit not only from the teaching, but also from contact with men of similar status who are studying subjects such as aircraft design, propulsion, electronics and aerodynamics.

The course will consist of lectures, laboratory demonstrations and individual research projects carried out under the supervision of members of the staff. Laboratory equipment will include test facilities for engines of all types, automatic transmission systems, brakes, tyres, radiators and shock absorbers. Handling qualities of full scale vehicles will be measured and the results so obtained compared with the theoretical performance.

Each student will select a subject for investigation, which should include both theoretical analysis and experimental research. These subjects will be chosen in consultation with the staff, who will take into account the individual requirements of each student and bear in mind his experience and probable future specialization. The subjects of the lectures will include mathematics, digital and analogue computers, materials, vibrations, mechanics, applied elasticity, thermodynamics, fuel technology, lubrication, economics of production, the human operator, safety considerations, vehicle stability and control, vehicle structural design, engine design, transmission design and experimental methods. Students who complete the course and submit theses judged to be of satisfactory quality will be awarded the Diploma in Automobile Engineering, D.Au.E. Suitable theses will be published.

The standard of entry is that of a university graduate or the holder of a Diploma in Technology. Candidates with other technical qualifications, including the Higher National Diploma or Certificate, will be considered for admission on the results of an interview or, in certain cases, an examination: two years' practical experience in industry is also normally required. The course will take place during the Autumn, Spring and Summer Terms of the academic year, and the dates of the terms for 1960-61 are respectively: 10th October to 16th December 1960, 16th January to 24th March 1961, and 24th April to 7th July 1961. For students who normally reside in the United Kingdom the tuition fee is £75, payable in advance, and the residence charge is at present £126 10s. This includes the cost of full board during the three terms, the rent of an individual study-bedroom, and the annual subscription to the Students' Society. Students accepted for the course are eligible to make application for a grant from the Bursary Fund if they are not already financially supported by a sponsoring firm or institution. The school is administered by The College of Aeronautics, and candidates should apply for admission, preferably before 1st August 1960, to the Warden, The College of Aeronautics, Cranfield, Bucks.

Russian Publication

ONE of Russia's leading technical monthly journals, *Vestnik Mashinostroeniya*, is now available in fully translated form under the title of *Russian Engineering Journal*. The translation and publication is done by the Production Engineering Research Association of Great Britain. Each issue contains detailed information on current Russian developments in the fields of design, construction and manufacture. An indication of the wide field covered is given by the following brief selection from recent issues: automatic handling, tool wear in fine turning, ultrasonic machining, die-casting machines, power transmitted by narrow vee-belts, plating magnesium alloys, electrolytic polishing, precision finishing of holes by ball rolling, pneumatic chucks, automation in car assembly, and high-frequency metallization. Details of the subscription rates for *Russian Engineering Journal* are obtainable from the Information Manager of the Production Engineering Research Association, at Melton Mowbray, Leicestershire.

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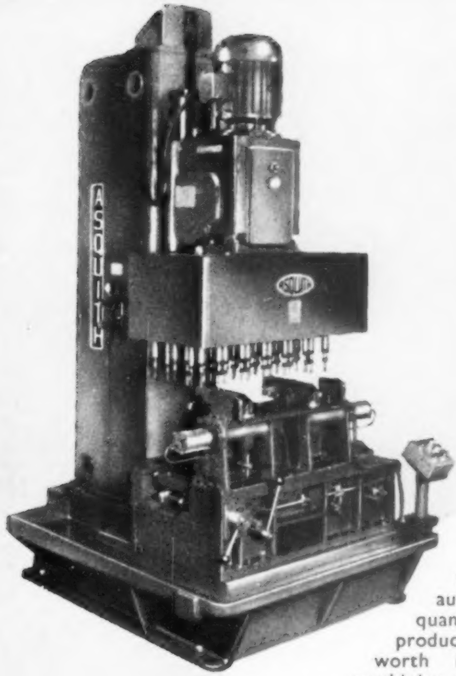

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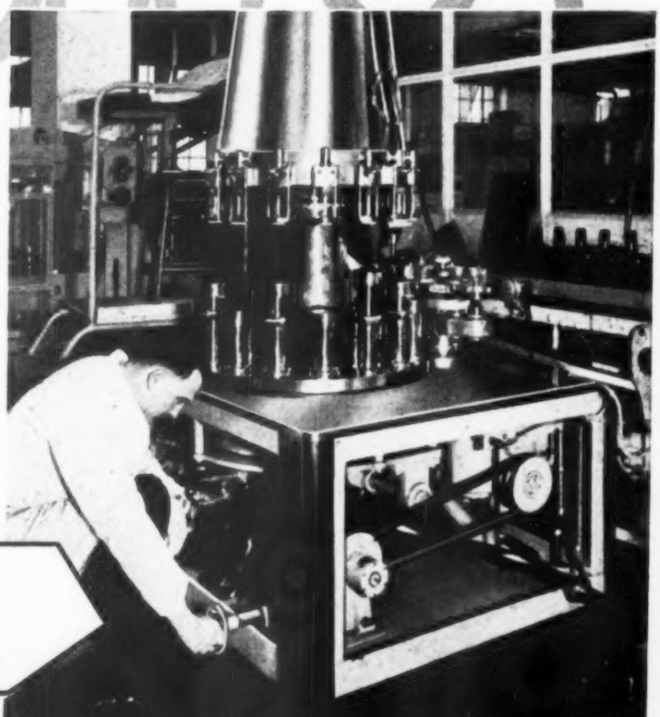

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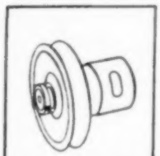

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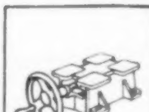

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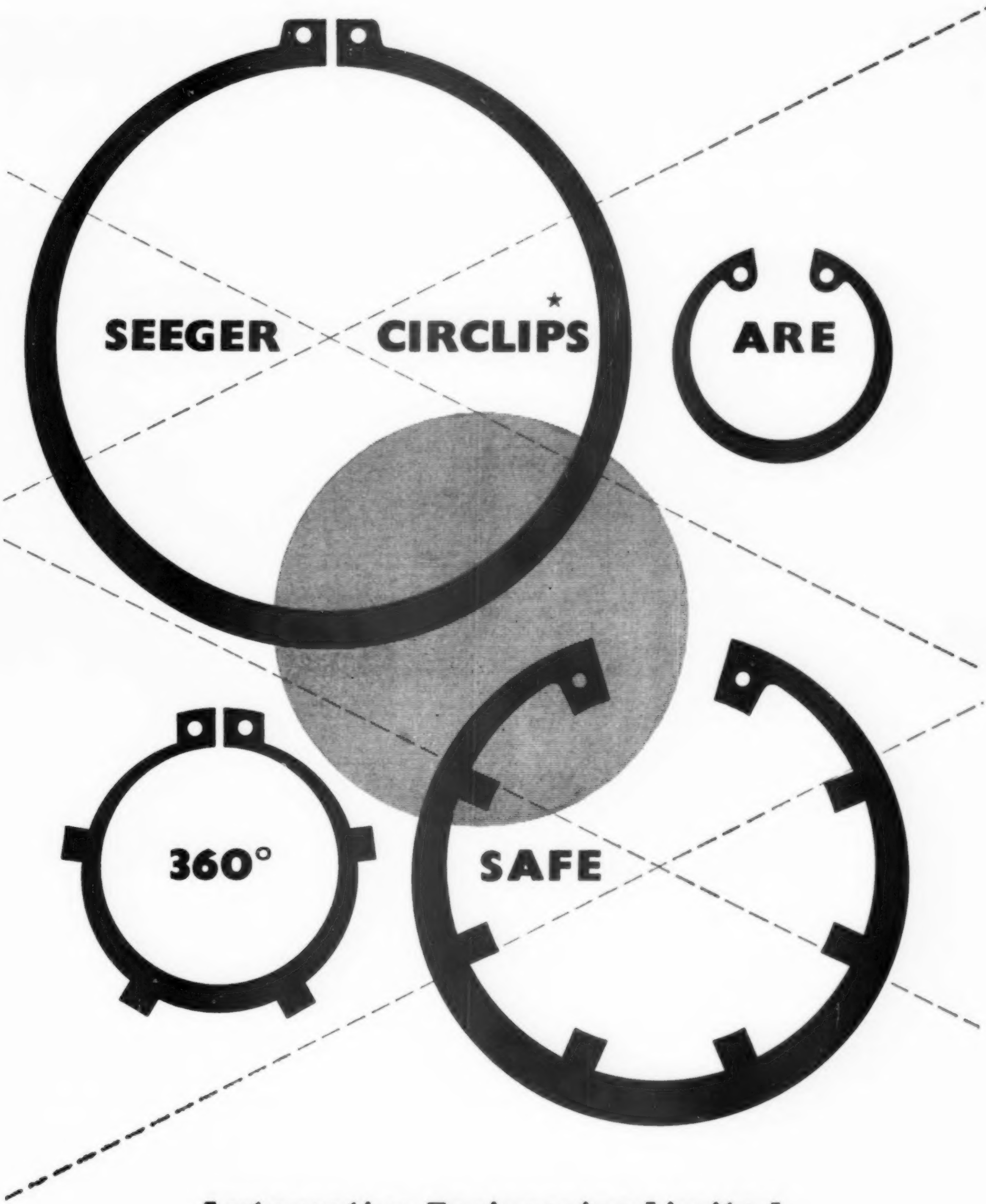
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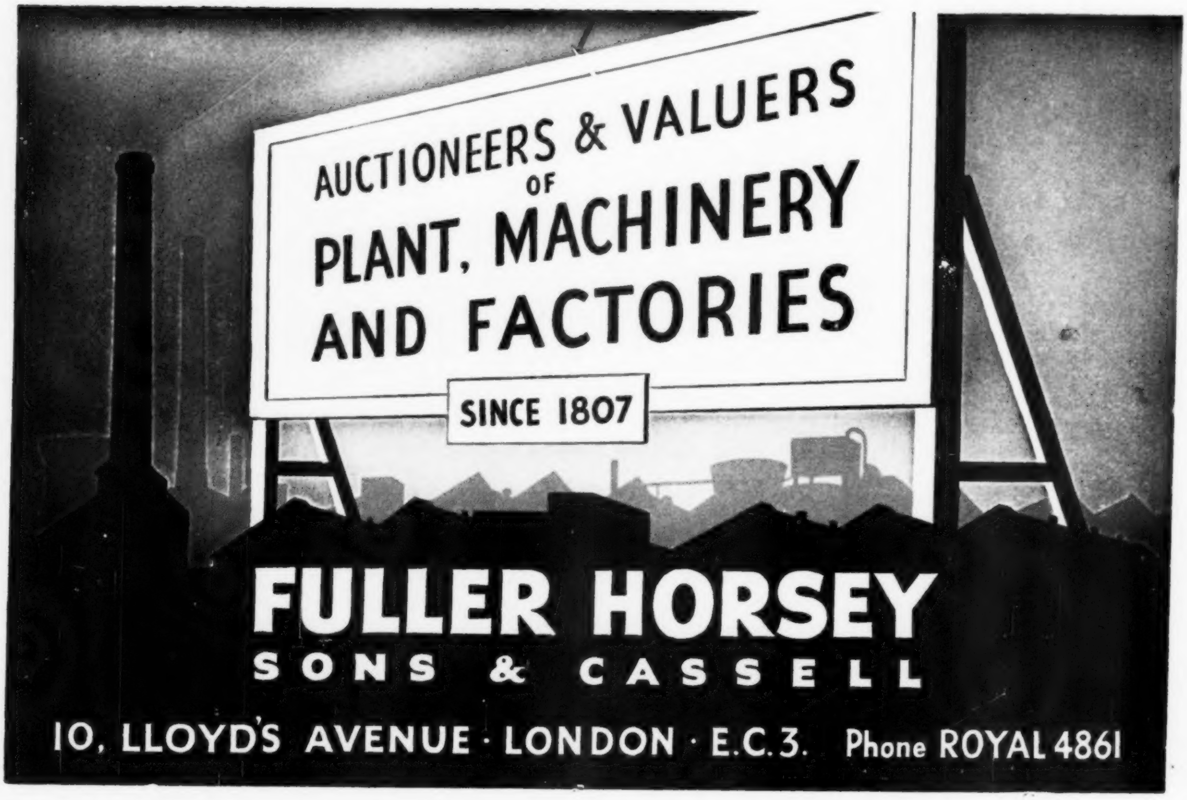
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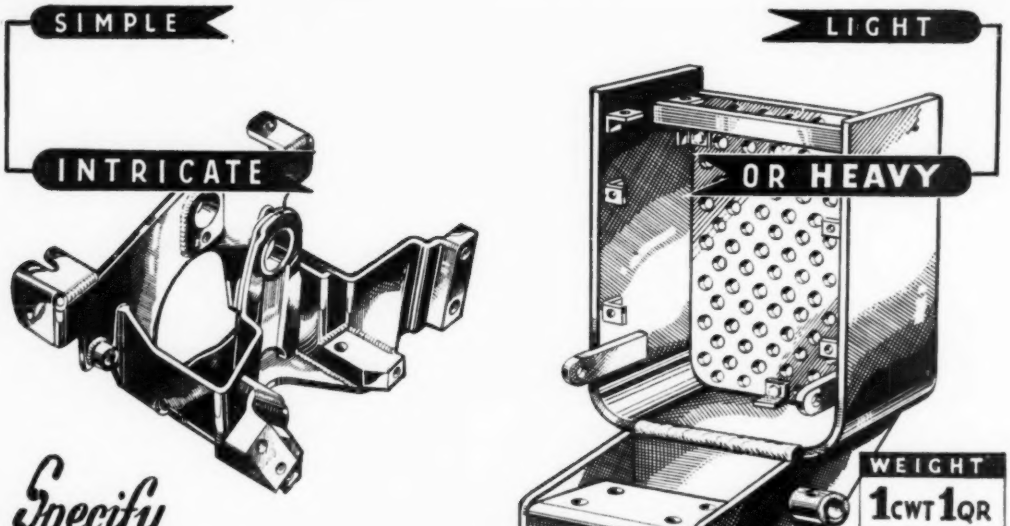
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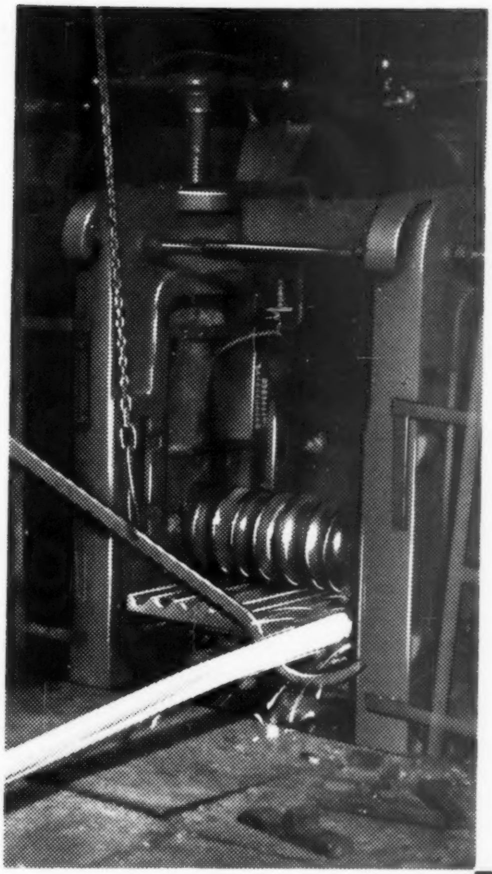

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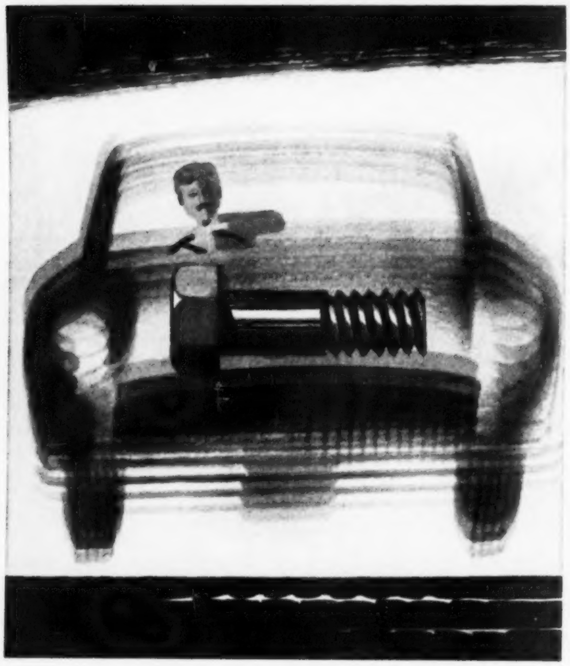

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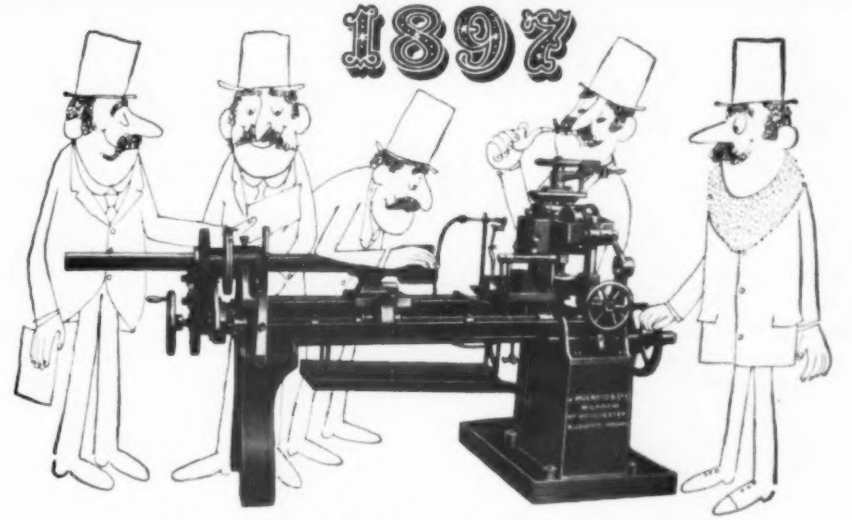

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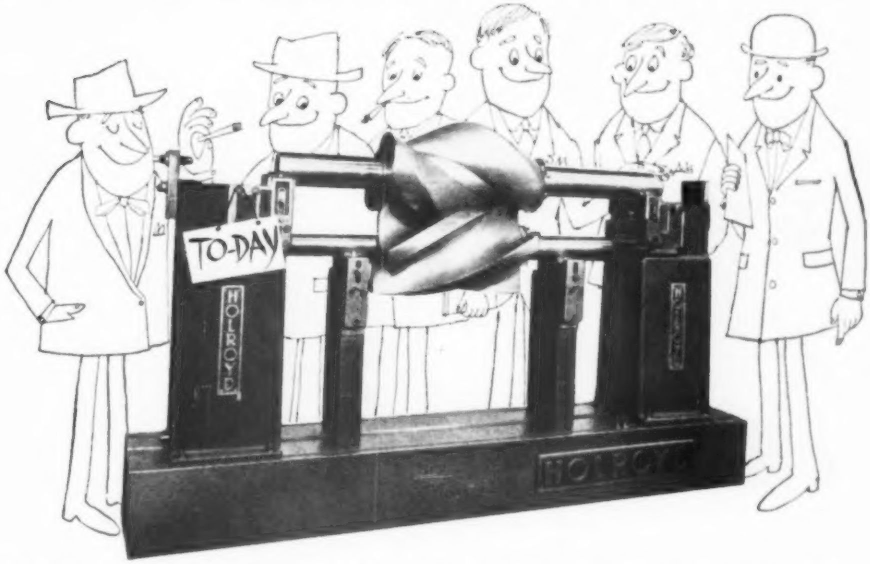

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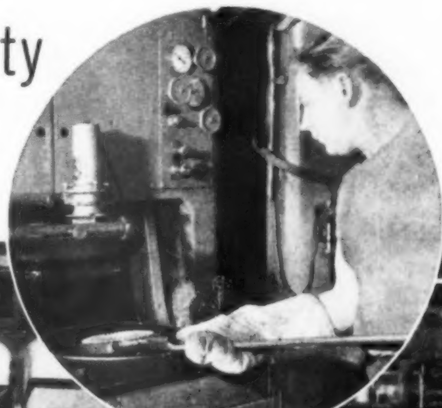

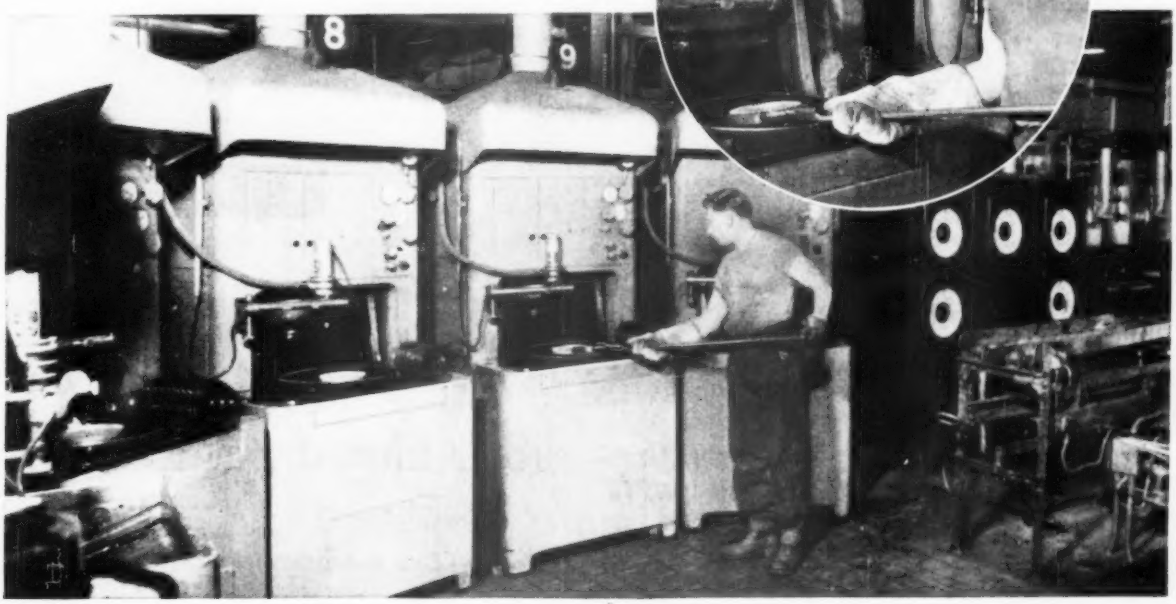

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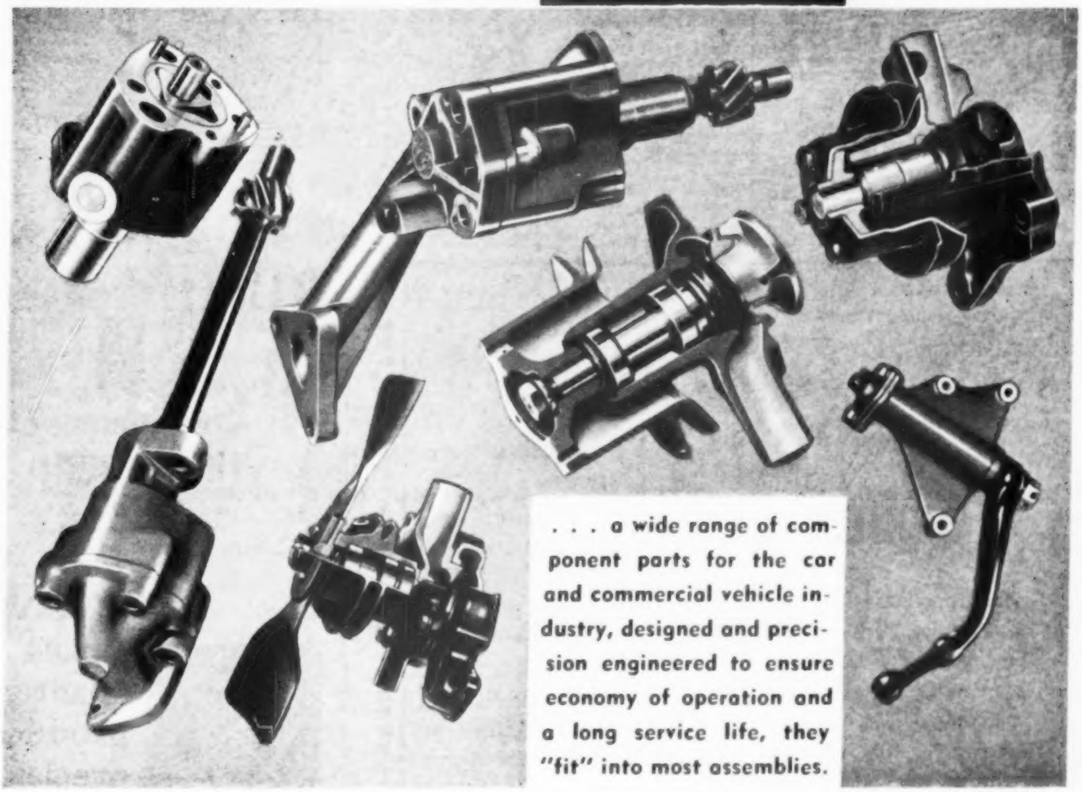

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